THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 45

MONTREAL MEETING
NEW YORK MEETING
1923

NEW YORK
PUBLISHED BY THE SOCIETY
29 WEST 39th STREET
1924

ERRATA

Attention has been called to the following errors and omissions in recent volumes of TRANSACTIONS:

VOLUME 31

Page 687, end of Paragraph 7, read Moisture loss = 0.65 [(212 — 100) + 966 + 0.48 (500 — 212)] = 660 B. t. u." for "Moisture loss = 0.65 (212 — 100) + 966 + 0.48 (500 — 212) = 660 B. t. u."

Page 687, Paragraph 8, Line 5, read "790" for "660" and "2535" for "2665" B. t. u.

Page 687, Paragraph 9, Line 4, read "2535" for "2665" and "5399" for "5676" B. t. u.

Page 687, Paragraph 9, Line 5, read "0.40" for "0.42" ton coal.

Page 702, Paragraph 50, Line 4, read "2500" for "2665" B. t. u.

Page 702, Paragraph 51, Line 3, read "5399" for "5676" B. t. u.

Page 703, Line 2, read "0.40" for "0.42" ton coal.

VOLUME 42

Page 347, read "Member of the Society" for "Non-Member."

VOLUME 43

Page 1239, footnote, read "See paper No. 1794" instead of "No. 1791."

VOLUME 44

Page 5, under Textile Machinery Session, Tuesday Morning, insert "Maintenance of Textile Machinery, Edwin H. Marble, published in *Mechanical Engineering,* May, 1922."

Page 545, Line 6, read "shaft" for "shift."

Page 545, Line 8, read "disturb it slightly" for "disturb slightly."

Index: "Properties of Metals at High Temperatures" (page 1130) and "Cost of Piping Materials" (page 1149), by W. S. Morrison (forming Appendices 1 and 2 to Paper No. 1874, by Geo. A. Orrok) were not indexed.

CONTENTS OF VOLUME 45[1]

Montreal and New York Meetings

[1] The Society shall not be responsible for statements or opinions advanced in papers or in discussions at meetings of the Society or of its Divisions or Sections, or printed in its publications. (B2 Par. 3).

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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Note: — Numbers in parentheses indicate the number of years the member has yet to serve.

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R. J. S. Pigott

Terms expire 1926

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Paul Diserens
C. E. Lucke
E. F. Miller
G. A. Orrok

Terms expire 1927

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* Members of the Executive Committee.

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The Regular Nominating Committee for Officers for 1922–1923 was nominated by the Conference of Local Section Delegates assembled at the 1921 Annual Meeting and confirmed by the Society at the Business Session at the same meeting.

Group	*Committee*
1 Boston, Worcester, Providence, Connecticut.	A. W. Honywill, Jr., *Hartford, Conn.* *Alternate* — E. H. Reed, *Worcester, Mass.*

Note: — Numbers in parentheses indicate the number of years the member has yet to serve.

2 New York, (Metropolitan Section).	H. H. BARNES, *New York.* *Alternate* — G. I. RHODES, *New York.*
3 Philadelphia, Lehigh Valley, Baltimore, Washington, D.C., Virginia, Atlanta, Birmingham, Central Pennsylvania, Plainfield.	W. W. VARNEY, *Chairman, Baltimore, Md.* *Alternate* — H. E. BUSSEY, *Atlanta, Ga.*
4 Eastern New York, Utica, Rochester, Syracuse, Buffalo, Ontario, Erie, Akron, Cleveland, Toledo, Columbus, Pittsburgh.	C. W. BENNETT, *Pittsburgh, Pa.* *Alternate* — R. W. ANGUS, *Toronto, Can.*
5 Chicago, Detroit, Milwaukee, Minneapolis, St. Paul, Indianapolis, Cincinnati, Tri-Cities.	PAUL DOTY, *St. Paul, Minn.* *Alternate* — W. A. HANLEY, *Indianapolis, Ind.*
6 St. Louis, New Orleans, Houston, Mid-Continent, Colorado, Kansas City.	V. J. AZBE, *Secretary, St. Louis.* *Alternate* — L. D. CRAIN, *Denver, Colo.*
7 Western Washington, Inland Empire, Oregon, Los Angeles, San Francisco.	F. S. KING, *Hollywood, Cal.* *Alternate* — R. L. DAUGHERTY, *Los Angeles, Cal.*

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AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE, SECTION ON ENGINEERING

ALEX. C. HUMPHREYS — V. E. MUNCY

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A. S. M. E. Representatives

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A. S. M. E. Delegates on American Engineering Council *

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Terms expiring Dec. 31, 1923	*Terms expiring Dec. 31, 1924*
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CHAS. A. BOOTH	M. E. COOLEY
GANO DUNN	E. R. FISH
H. H. ESSELSTYN	ARTHUR M. GREENE, JR.
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* The Governing Board of the Federated American Engineering Societies.

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D. S. Jacobus
Term expires Feb., 1924

J. H. Barr
Term expires Feb., 1925

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Term expires Feb., 1924

Walter M. McFarland
Term expires Feb., 1925

Ambrose Swasey
Term expires Feb., 1926

Fred J. Miller
Term expires Feb., 1927

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Term expires June 30, 1923

D. S. Jacobus
Term expires June 30, 1924

Albert Kingsbury
Term expires June 30, 1925

R. J. S. Pigott
Term expires June 30, 1926

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Term expires Jan., 1924

W. L. Saunders
Term expires Jan., 1925

W. F. M. Goss
Term expires Jan., 1926

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Charles Russ Richards

Herbert S. Philbrick

JOSEPH A. HOLMES MEMORIAL BOARD

William H. Bixby

SUMMARY OF MEMBERSHIP BY RESIDENCE

United States and Possessions

Alabama	103	Nebraska	28
Alaska	2	Nevada	3
Arizona	26	New Hampshire	49
Arkansas	12	New Jersey	1148
California	689	New Mexico	9
Canal Zone	5	New York	4483
Colorado	74	North Carolina	81
Connecticut	743	North Dakota	5
Delaware	88	Ohio	1226
District of Columbia	186	Oklahoma	87
Florida	52	Oregon	44
Georgia	124	Pennsylvania	1868
Hawaiian Islands	22	Philippine Islands	24
Idaho	3	Porto Rico	13
Illinois	1078	Rhode Island	156
Indiana	263	South Carolina	28
Iowa	46	South Dakota	4
Kansas	55	Tennessee	107
Kentucky	61	Texas	161
Louisiana	114	Utah	19
Maine	45	Vermont	34
Maryland	213	Virginia	119
Massachusetts	1359	Washington	104
Michigan	568	West Virginia	61
Minnesota	133	Wisconsin	338
Mississippi	10	Wyoming	12
Missouri	317		
Montana	16	Total	16618

Foreign Countries

North America		**West Indies**	
Canada	244	Cuba	65
Newfoundland	1	Jamaica	2
Mexico	33	Santo Domingo	4
	278	Trinidad	1
			72
Central America		**South America**	
Costa Rica	4	Argentina	19
Guatemala	1	Bolivia	2
Honduras	1	Brazil	13
	6		

South America (*continued*)	
Chile	24
Colombia	2
Peru	4
Uruguay	2
Venezuela	2
	68

Africa	
Cape Colony	2
Egypt	1
Natal	1
Senegal	1
Transvaal	9
	14

Asia	
China	32
Dutch East Indies	2
India	29
Japan	27
Palestine	1
Persia	1
Singapore Straits Settlements	2
	93

Australasia	
Australia	16
Tasmania	1
	17

Europe	
Belgium	3
Czechoslovakia	5
Denmark	10
Finland	3
France	35
Germany	19
Great Britain	96
Greece	1
Holland	4
Italy	6
Norway	4
Poland	1
Roumania	3
Spain	7
Sweden	15
Switzerland	9
Turkey	1
	222
Total in Foreign Countries	770

Membership in United States	16618
Membership in Foreign Countries	770
Present Address Unknown	64
Total Membership	17452

Summary of Membership by Grades

Honorary Members	19
Members	7886
Associates	831
Associate-Members	4260
Juniors	4456
Total	17452

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 45—1923

THE activities of the Society for 1923 are recorded in Volume 45. In this volume will be found papers and discussions presented at the Spring Meeting held at Montreal and at the Annual Meeting held in New York. In addition are other papers which the Committee has deemed of sufficient interest to include in Transactions.

In selecting material for Transactions the Committee is guided by the principle of choosing material of permanent and technical value. For this reason many discussions are condensed and some papers and addresses of less technical interest are given in abstract form. In such cases more complete forms of the papers may be found in MECHANICAL ENGINEERING.

Attention is particularly called to the fact that all papers presented before general meetings of the Society are mentioned in the index to Transactions. Where the papers have not been printed in this volume of Transactions reference is made to the issue of MECHANICAL ENGINEERING in which they appeared. The papers presented before Local Sections that were published in MECHANICAL ENGINEERING, as well as other important contributions to it, may be found by reference to the index in this volume.

The Transactions also contains a statement of the reports of technical committees and codes published during the year, and the index tells where they may be found in printed form.

JOHN LYLE HARRINGTON

John Lyle Harrington was born in Lawrence, Kansas, December 7, 1868. He was educated in the schools of Lawrence and at the University of Kansas, from which he was graduated in 1895 with the degrees of Bachelor of Arts, Bachelor of Science, and the graduate degree of Civil Engineer, receiving the highest honors of his class (commencement appointment and election to Sigma

Xi). Later he received the degree of Master of Science from McGill University, Montreal.

His summer vacations prior to graduation and nine months after graduation were spent with J. A. L. Waddell, Consulting Engineer, Kansas City, Mo. He then served successively in the engineering departments of the Elmira Bridge Co., Elmira, N. Y., the Pencoyd Iron Works, Philadelphia, and the Keystone Bridge Works of the Carnegie Steel Company at Pittsburgh, until January, 1898. Following that date he designed and organized the operation of the structural shops of the Cambria Steel Company at Johnstown, Pa., and served successively as assistant chief engineer of the Bucyrus Company, South Milwaukee, Wis.; assistant chief engineer of the Northwestern Elevated Railway Company of Chicago, in charge of detailing, fabrication, and inspection in the shops; designer for the Berlin Iron Bridge Company, East Berlin, Conn., and assistant engineer of bridges and buildings of the Baltimore & Ohio Railway.

From 1901 to 1905 Mr. Harrington was executive engineer of the C. W. Hunt Company, hoisting and conveying machinery builders of New York; 1905 to 1906, chief engineer and manager of the Locomotive and Machine Company of Montreal, a subsidiary of the American Locomotive Company; 1907 to 1914, inclusive, partner in the firm of Waddell and Harrington, Consulting Engineers, Kansas City, Mo., engaged chiefly upon the design of important bridges, and making a specialty of movable spans on which he has taken out, in his firm's name or in his own, a number of important patents; 1915 to date, senior partner in the firm of Harrington, Howard and Ash, Consulting Engineers, Kansas City, Mo., designing important bridges, also heavy drainage work and similar lines of construction. During the fifteen years prior to 1924 he was responsible for the design and construction of approximately one hundred million dollars' worth of heavy bridge work, including about one hundred movable spans.

Mr. Harrington became a member of the Society in 1903 and served as its Vice-President in 1921–1922 and President in 1922–1923. He is also a member of the American Society of Civil Engineers, the Institution of Civil Engineers, London, the Engineering Institute of Canada, the American Railway Engineering Association, the American Society for Testing Materials, the honorary societies Sigma Xi and Tau Beta Pi (Past-President), the Engineers' Club of Kansas City (Past-President), the University Club of Kansas City, and other clubs.

No. 1885

SOCIETY AFFAIRS

THE SPRING MEETING

Montreal, Canada, May 28 to 31

THE hearty coöperation of the Engineering Institute of Canada in the 1923 Spring meeting of The American Society of Mechanical Engineers was a splendid exemplification of the internationality of engineering interests. Out of a total registration of 631 at the meeting, which was held in Montreal, with headquarters at the Mount Royal Hotel, 149 were members of the Engineering Institute of Canada. Seven of the seventeen papers presented were contributed by Canadian engineers, and Canadian hospitality was evidenced on all occasions.

Nine past-presidents attended the Council Meeting on the opening morning of the meeting, at which all but four of its members were present.

A conference of Local Sections' Delegates was also held Monday morning. At the Business Meeting Monday afternoon Walter J. Francis, president of the Engineering Institute of Canada, extended a greeting on behalf of the engineers of Canada, to which President Harrington responded. The selection of Cleveland as the place for the 1924 Spring Meeting was ratified, and John Price Jackson spoke concerning the proposed Sesquicentennial Exposition in Philadelphia in 1926. Secretary Calvin W. Rice then presented his travelogue on South America.

The Code on Internal-Combustion Engines and the first two chapters of the Code on Instruments and Apparatus were presented and discussed at a Public Hearing on Monday afternoon; the Proposed Rules for the Care of Power Boilers, and the Proposed Rules for Inspection of Materials for Boilers on Tuesday and Wednesday afternoons, respectively; and the report of the Research Sub-Committee on Fluid Meters at an open meeting on Thursday afternoon. The regular May Meeting of the A.S.M.E. Boiler Code Committee was held on Thursday also, and was attended not only by a large number of the members, but also by inspectors from the Canadian provinces and boiler manufacturers and engineers from various points who are interested in the Committee's activities.

The technical program opened Tuesday morning with simulta-

neous sessions on Power, Management, and Railroads. A second session on Power, and sessions on Port Development and Textiles, occurred on Wednesday morning, and on Thursday morning Fuels, Machine-Shop Practice, and General Sessions were held.

The social events of the meeting included an informal bridge for the ladies and a smoking concert for the men on Tuesday evening, and a dinner dance Wednesday evening. On Monday evening the A.S.M.E. Council was entertained at dinner by the Council of the Engineering Institute of Canada.

Large groups visited the industrial plants that were thrown open for inspection during the meeting. The Canadian Pacific Railway conveyed about a hundred to their Angus Shops on Tuesday afternoon, where locomotives and freight and passenger cars are manufactured. The same afternoon another large group journeyed to the Cedars Rapids Plant of the Montreal Light, Heat and Power Company.

On Wednesday the main body of members visited the port of Montreal. The ladies were taken about in the Harbor Commissioner's boat and the other members of the party inspected the grain elevators, cold-storage warehouses, and other port facilities. On the same afternoon the various Montreal breweries were visited.

Over 100 availed themselves of the opportunity offered Thursday afternoon to visit the Dominion Engineering Works, where a number of large hydraulic turbines and large paper machines were under construction. A turbine-runner casting was poured in the foundry during the inspection. Following the inspection, the special train furnished by the courtesy of the Canadian National Railways took the party to the Lachine Wharf where the steamer *Empress* was boarded for a trip through the Lachine Rapids back to the city.

Also on Thursday afternoon a group of about 50 visited McGill University, saw a demonstration of the oscilloscope, and were then conducted through the historic halls. On Thursday night about 50 members of the Society boarded two special cars which conveyed them to the St. Maurice Valley. At Grand Mere the party inspected the large paper mill of the Laurentide Company, and at Shawinigan Falls visited the Canada Carbide Company's plant and the hydroelectric plant of the Shawinigan Water and Power Company.

Many of the 119 ladies registered at the meeting participated in the excursions. Other groups visited the famous fur stores of Montreal and scenic and historical points.

The local executive committee consisted of H. H. Vaughan, Chairman; Major J. A. Duchastel, Vice-Chairman; Fraser S. Keith, Secretary; and Fred B. Brown, John T. Farmer, George R. MacLeod, and Major C. M. McKergow. The chairmen of the sub-committees were H. H. Vaughan, Finance; Fred B. Brown, Entertainment; Major C. M. McKergow, Hotels and Information; Major J. A. Duchastel, Excursions (with H. O. Keay and R. W

Angus on Local Excursions); John T. Farmer, Technical and Professional; J. H. Hunter, Reception; Fraser S. Keith, Printing and Publicity; and Mrs. Fred B. Brown, Ladies'.

PROGRAM

Monday Morning, May 28

Opening of Headquarters and Registration Bureau.
Council Meeting.
Conference of Local Sections' Delegates.

Monday Afternoon

BUSINESS MEETING

Business from the Council; business from the membership; report from American Engineering Council by President Harrington; address on Plan and Scope of Sesquicentennial Celebration in Philadelphia, John Price Jackson.

Travelogue on South America, Calvin W. Rice.

PUBLIC HEARING

Power Test Code for Instruments and Apparatus, Chapters 1 and 2, Power Test Code for Internal-Combustion Engines.

Monday Evening

Joint Dinner, Council of Engineering Institute of Canada and Council of A.S.M.E.

Tuesday Morning, May 29

SIMULTANEOUS SESSIONS

POWER (1)

Power Development in the Province of Quebec, Julian C. Smith.
The Development of Hydroelectric Power Plants in Ontario, Frederick A. Gaby.

MANAGEMENT

Management Engineering in the Paper Industry, R. B. Wolf.[1]
A Practical Laboratory and Drawing-Room Course in Industrial Engineering at Cornell University, Myron A. Lee.[2]

RAILROAD

Steel-Car Construction at the Angus Shops of the Canadian Pacific Railway, H. R. Naylor.
Recent Developments of the Motor Coach, C. E. Brooks.[2]

Tuesday Afternoon

Public Hearing: Proposed Rules for the Care of Steam Boilers in Service.

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[1] Published in Mechanical Engineering, May, 1923.
[2] Published in Mechanical Engineering, August, 1923.

Tuesday Evening

Informal Bridge Party for Ladies.
Smoking Concert.

Wednesday Morning, May 30

SIMULTANEOUS SESSIONS

POWER (2)

Modern Hydraulic Turbines of Large Capacity, H. G. Acres.
Sectionalization and Remote Control of High-Pressure Steam Lines, Peter Payne Dean.[1]

PORT DEVELOPMENT

The Port of Montreal, Frederick W. Cowie.[2]
Material-Handling Problems in Pier Design, Carroll R. Thompson.[2]
Report of Committee on Formula.

TEXTILES

Bleachery Engineering and Operation, F. P. Bascom and J. C. McDowell.

Wednesday Evening

Dinner Dance.

Thursday Morning, May 31

SIMULTANEOUS SESSIONS

FUELS

Determination of Chimney Sizes, Alfred Cotton.[2]
Lignite Char: Its Production and Possibilities, O. P. Hood.

MACHINE-SHOP PRACTICE

Recent Developments in Balancing Machines, C. R. Soderberg.
Symposium on Machine-Tool and Paper Industry:
  Introduction by G. E. Williamson.
  Shop Problems in Paper-Machine Manufacturing, E. T. Spidy.
  General Requirements of Machine Tools for Paper Machinery Manufacture, Geo. S. Barton.
  General Requirements of Machine Tools for Pulp and Paper Machinery Manufacture, H. L. Kutter.
Plant Equipment Necessary for Producing Paper-Working Machinery, Jas. A. Cameron.
Machine-Tool Requirements of Pulp and Paper Mills, E. B. Wardle.
Pulp and Paper Mill Repair Shop, Edw. Hutchins.

GENERAL

The Control of Civil Aviation, J. A. Wilson.[3]

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[1] Published in Mechanical Engineering, August, 1923.
[2] Published in Mechanical Engineering, September, 1923.
[3] Published in Mechanical Engineering, October, 1923.

Endurance Test Data and Their Interpretation, K. Heindlhofer and H. Sjövall.

Bending Stresses in Curved Tubes of Rectangular Cross-Section, S. Timoshenko.

THE ANNUAL MEETING

New York, N. Y., December 3 to 6

As diversified in interest as its predecessors but less crowded, the program of the Forty-Fourth Annual Meeting was the most satisfactory of any in recent years. The technical sessions, eighteen in number, dealt largely with the principles of mechanical engineering as applied to the problems of industry, and the excursions and the Second National Exposition of Power and Mechanical Engineering, which paralleled the meeting, offered opportunities to see the actual application of some of the principles.

The registration at the meeting was 1852. As usual, the first day was devoted to meetings of the Council and Local Sections' delegates. In addition to the transaction of routine business the Council passed a resolution in favor of the reduction of taxes on earned incomes, endorsing the recommendations of Secretary Mellon and urging upon Congress the amendment to the Internal Revenue Act proposed by him. Chief among the Society problems which came up for consideration was the advisability of increasing membership dues. The Policy Report and financial problems were discussed from all angles and on Tuesday afternoon, at the Business Meeting, it was voted to refer the matter to the members by letter ballot.

The following awards were made at the Business Meeting: Life Membership to Prof. John Airey, Ann Arbor, Mich.; Junior Prize to S. S. Sanford, Detroit, Mich.; and Student Prizes to Charles F. Olmstead, Minneapolis, Minn., and H. E. Doolittle, San Diego, Cal. The following standards were read by title: Standards for Transmission Chains and Sprockets, including those on Revised Sprocket Tooth Form, Space and Straddle Cutters for Sprocket Teeth, and Roller-Chain Nomenclature; and Standards for Screw Threads for Bolts, Machine Screws, Commercial Tapped Holes, etc.

Following the presidential address of John Lyle Harrington on Monday evening, the 1922 Society Medal was presented to Frederick A. Halsey for his invention of the premium system of wage payments and the great improvement resulting therefrom in the relations between employers and workmen. The 1923 Medal was awarded to Past-President John R. Freeman for his eminent service to engineering and manufacturing by his meritorious work in fire prevention and the preservation of property.

The new officers, whose election was announced following the presidential address on Monday evening, were installed at the first

meeting of the new Council on Friday morning. They were President Fred R. Low; Vice-Presidents George I. Rockwood, W. J. Sando, and H. Birchard Taylor; Managers E. O. Eastwood, E. R. Fish, and Frank A. Scott; and Treasurer William H. Wiley. The new Council approved the holding of a Regional Meeting in Colorado in May or June, 1924, and transacted other routine business.

Two of the technical sessions were held in coöperation with other societies: one on Tuesday morning with the American Society of Refrigerating Engineers, and one on Wednesday evening with the American Society of Civil Engineers and the American Institute of Electrical Engineers. John R. Freeman, Past-President of both the A.S.C.E. and the A.S.M.E., was the chief speaker at the latter, which was devoted to the subject of hydroelectric development.

One of the best-attended sessions was that on steam power, held on Thursday under the auspices of the Power Division. Over 700 were in attendance during the morning, when six valuable papers were presented, and a large number remained through the afternoon to participate in the discussion.

Other sessions under the auspices of the professional divisions of the Society dealt with textiles, coal storage, railroads, ordnance, forest products, machine-shop practice, aeronautics, management, oil engines, and water measurement.

One general session was held, and two public hearings on codes. The first of these was on Part II of the Report of the Special Committee on Fluid Meters, and the second on the Power Test Codes for Stationary Steam-Generating Units and for Locomotives. A conference on education and training for the industries was also held, at which there was a discussion of education of non-college grade for the industries, and an important session on steam-table research. Many committee meetings and conferences took place during the week. The Student Branch session on Wednesday was attended by representatives of twenty-eight branches.

In addition to the annual reception on Monday evening the social program included a smoker and dinner for the men and a benefit card party for the ladies on Tuesday; a ladies' reception and tea on Wednesday afternoon; and the annual dinner dance on Thursday evening. The proceeds of the card party were contributed to the Scholarship Fund of the Women's Auxiliary. Three members, Messrs. Odell, Lyne, and Grimshaw, who were present at the organization meeting of the Society in 1880, attended the smoker, at which the total attendance was 520. The alumni of various technical colleges held reunions Friday evening following the meeting.

The program for the ladies included trips to the new store of A. A. Vantine & Co.; the plant of the Ward Baking Company on East 143d St.; the Kirkman Soap Factory, Brooklyn; the U. S. S. *Colorado;* Good Housekeeping Institute and the Studio of Furnishings and Decorations; the Colgate & Co. plant at Jersey City;

galleries of Senator William A. Clark; Aeolian Hall; and Broadcasting Station WJZ. A business meeting of the Women's Auxiliary was held Thursday afternoon.

Technical excursions were made by members to the American Machine & Foundry Co., Brooklyn; Wright Aeronautical Corporation, Paterson, N. J.; the factory of the E. W. Bliss Company, Brooklyn, which specializes in pressed-metal machinery, automatic can machinery, dies, special machinery for stampings, drop forgings, etc.; the working model of the moving rapid-transit platform proposed for 42d St. subway traffic; the Star Automobile Factory of the Durant Motor Company at North Elizabeth, N. J.; and the Tidewater Oil Co. refining plant at Bayonne, N. J.

PROGRAM

Monday Morning and Afternoon, December 3

Council Meeting.
Conference of Local Sections' Delegates.

Monday Evening

Presidential Address, John Lyle Harrington.
Presentation of A.S.M.E. Medals.
Report of Tellers of Election of Officers.
Introduction of President-Elect Low.
Reception.

Tuesday Morning, December 4

SIMULTANEOUS SESSIONS

JOINT SESSION WITH AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

Heat Transfer for Water Flowing Inside Pipes, W. H. McAdams and T. H. Frost.

The Economical Thickness of Insulation in Refrigerator Cars, Arthur J. Wood and Philip X. Rice.

TEXTILES

Organization and Construction of Woolen Mills, A. W. Benoit.[1]

A Steam-Loss Prevention Plan Operating in a Textile-Finishing Plant, H. M. Burke.

FLUID METERS

Open Meeting for discussion of Part II of the Report of the Special Committee on Fluid Meters.

GENERAL

The Bending and Torsion of Multi-Throw Crankshafts on Many Supports, S. Timoshenko.

A Graphical Study of Journal Lubrication, H. A. S. Howarth.

Stress Distribution in Rotating Gear Pinions as Determined by the Photoelastic Method, Paul Heymans and A. L. Kimball, Jr.[2]

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[1] Published in Mechanical Engineering, April, 1924.
[2] Published in Mechanical Engineering, March, 1924.

Tuesday Afternoon

BUSINESS MEETING

Annual Report of Council; election of Regular Nominating Committee for 1924; presentation of Constitutional Amendment on increase in dues; presentation of awards; reading of standards by title.

PUBLIC HEARING

Power Test Code for Stationary Steam-Generating Units.
Power Test Code for Locomotives.

Tuesday Evening

Smoker, and Dinner.
Ladies' Benefit Card Party.

Wednesday Morning, December 5

SIMULTANEOUS SESSIONS

COAL STORAGE

Factors in the Spontaneous Combustion of Coal, O. P. Hood.[1]
Economic Phases of Coal Storage, F. G. Tryon and W. F. McKenney.[2]
Coal-Storage Systems, H. E. Birch and H. V. Coes.[3]

RAILROAD

Modern Subway Cars and Their Operation, Selby Haar.[4]

ORDNANCE

New Instruments for Physical Measurements, W. H. Tschappat.[1]
Some Production Problems in the War Department's Preparedness Program, H. W. Churchill.[5]

FOREST PRODUCTS

Reforestation and Timber Conservation, John W. Blodgett.[2]

Wednesday Afternoon

SIMULTANEOUS SESSIONS

STEAM TABLE RESEARCH

Report of Executive Committee, Steam Table Fund, Geo. A. Orrok.[2]
Summation of Research Results, Harvey N. Davis.[2]
Research Progress Reports, F. G. Keyes and R. Kleinschmidt.[2]
A Calorimetric Method of Surveying the Behavior of Steam, N. S. Osborne.[2]

EDUCATION AND TRAINING FOR THE INDUSTRIES

Introductory Statement, Dr. Ira N. Hollis.
Industrial Education Abroad, R. D. Sackett.

Conference of Student Branches.

Ladies' Tea and Reception.

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[1] Published in Mechanical Engineering, December, 1923.
[2] Published in Mechanical Engineering, February, 1924.
[3] Published in Mechanical Engineering, April, 1924.
[4] Published in Mechanical Engineering, November, 1923.
[5] Published in Mechanical Engineering, May, 1924.

Wednesday Evening

HYDROELECTRIC SESSION

THE PRINCIPLES UNDERLYING HYDROELECTRIC DEVELOPMENT, John R. Freeman.

THE STUDY OF HYDROELECTRIC POSSIBILITIES, John P. Hógan.

WATER-POWER COSTS VERSUS STEAM-POWER COSTS, Geo. A. Orrok.

THE INTERCONNECTION OF POWER SYSTEMS, Harold W. Buck.

Thursday Morning, December 6

SIMULTANEOUS SESSIONS

STEAM POWER

BOILER-PLANT ECONOMICS, N. E. Funk and F. C. Ralston.

BOILER-TEST RESULTS WITH PREHEATED AIR, COLFAX STATION, DUQUESNE LIGHT COMPANY, C. W. E. Clarke.

THE MARGINS OF POSSIBLE IMPROVEMENT IN THE CENTRAL-STATION STEAM PLANT, Ernest L. Robinson.

ECONOMY CHARACTERISTICS OF STAGE FEEDWATER HEATING BY EXTRACTION, E. H. Brown and M. K. Drewry.

REHEATING IN CENTRAL STATIONS, W. J. Wohlenberg.

HIGH PRESSURE, REHEATING, AND REGENERATING FOR STEAM POWER PLANTS, C. F. Hirshfeld and F. O. Ellenwood.

MACHINE-SHOP PRACTICE

PRESSED-METAL ENGINEERING: SOME PRINCIPLES AND EXAMPLES, Douglas P. Cook.[1]

THE DEVELOPMENT OF MODERN STAMPING PRACTICE, W. W. Galbreath and John R. Winter.[3]

PROGRESS REPORT ON THE PRESENT STATUS AND FUTURE PROBLEMS OF THE ART OF CUTTING METALS.[2]

AERONAUTICS

RESISTANCE OF VARIOUS ALUMINUM ALLOYS TO SALT-WATER CORROSION, D. Basch and M. F. Sayre.[3]

RECENT OBSERVATIONS REGARDING THE CORROSION, CLEANSING, AND PROTECTION OF ALUMINUM, Henry A. Gardner.[3]

THE COMMERCIAL POSSIBILITIES OF THE AIRPLANE, Archibald Black and D. R. Black.[1]

NIGHT-FLYING EQUIPMENT AND OPERATION, H. R. Harris and D. L. Bruner.[4]

Thursday Afternoon

SIMULTANEOUS SESSIONS

MANAGEMENT

THE MECHANICAL ENGINEERING OF MANAGEMENT IN THE METAL-WORKING TRADES, Robert T. Kent.[4]

THE MECHANICAL ENGINEER IN THE MANAGEMENT OF WOODWORKING INDUSTRIES, W. D. Churchill.[5]

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[1] Published in MECHANICAL ENGINEERING, March, 1924.
[2] Published in MECHANICAL ENGINEERING, January, 1924.
[3] Published in MECHANICAL ENGINEERING, April, 1924.
[4] Published in MECHANICAL ENGINEERING, May, 1924.
[5] Published in MECHANICAL ENGINEERING, December, 1923.

The Relation of Mechanical Engineering To Management in the Textile Industry, E. H. McKitterick.[1]

OIL-ENGINE

The Solid-Injection Oil Engine, H. F. Shepherd.
The Economic Status of the Diesel Engine, L. H. Morrison

WATER MEASUREMENT

Flow of Water in Short Pipes, O. W. Boston.
The Salt Velocity Method of Water Measurement, Charles M. Allen and Edwin A. Taylor.
The Gibson Method and Apparatus for Measuring the Flow of Water in Closed Conduits, Norman R. Gibson.

Business Meeting, Women's Auxiliary.

Thursday Evening

Dinner Dance.

[1] Published in Mechanical Engineering, June, 1924.

No. 1886

HYDROELECTRIC POSSIBILITIES OF QUEBEC[1]

BY JULIAN C. SMITH,[2] MONTREAL, CANADA
Non-Member

The three watersheds of the province of Quebec have an available water-power of 5,250,000 *kw. of continuous power, of which at present about* 800,000 *kw. have been developed and utilized, leaving available as commercial power about* 4,500,000 *kw. of continuous power, or* 12,000,000 *hp. at* 50 *per cent load factor. The load factor of the industries of the province range about as follows: paper mills,* 65 *to* 75 *per cent; carbide and electrochemical plants,* 85 *to* 90 *per cent; cement mills,* 80 *to* 90 *per cent; asbestos and mining industries,* 45 *per cent. At the present rate of increase in the use of electric power, the total available hydroelectric power in the province will be completely utilized within thirty years.*

THE PROVINCE of Quebec comprises that vast stretch of territory lying east of the Ottawa River and north of the United States boundary line, bounded on the north by Hudson Strait, on the west by Hudson Bay, and on the east by Labrador. It is 2000 miles in width from north to south and 1350 miles from east to west, having an area of 706,804 square miles, or almost 25 per cent of that of the United States. The northern section is for the most part uninhabited, but the southwestern portion lying along the lower part of the Ottawa and St. Lawrence Rivers, and extending south to the American border and east to the Saguenay River, is an important and rapidly growing industrial section.

2 The climate of Quebec is quite similar to that of northern New England. The winters are severe, and snow covers the ground from the middle of November to the end of March. The summers are warm and pleasant, and the precipitation, which is fairly evenly distributed over all seasons, averages about forty inches per year.

[1] For discussion, see p. 36.

[2] Vice-President and General Manager, Shawinigan Water and Power Co.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Slightly abridged.

The Watersheds of the Province

3 Topographically the country north of the St. Lawrence River is divided into three great watersheds. On the south side of the river there are no large water powers, as the watershed is comparatively narrow and, although well watered, is traversed by only a few rivers of any magnitude.

4 The three great watersheds lying to the north of the St. Lawrence River are the south watershed, draining into the St. Lawrence River; the north watershed, into the James and Hudson Bays; and the east watershed, through Labrador into the Atlantic Ocean. Of these the south watershed, through which flow the tributaries of the St. Lawrence from the Ottawa to the Notashkwan, is the most important. The height of land dividing the north and south watersheds lies some 200 miles north of the St. Lawrence and runs roughly parallel to it. On either side of this height of land there is a vast plateau, with a mean elevation of from 1200 to 2000 ft., which begins at the Ottawa with a width of some 150 miles and spreads out like a fan toward Labrador, where it attains a width of 500 miles or more. The southern edges of this plateau are for the most part near the St. Lawrence, so that the rivers which traverse it make their sharpest descent to sea level not far distant from the main river, and as a consequence much of the water power of this watershed is within easy reach of the industrial section of the province.

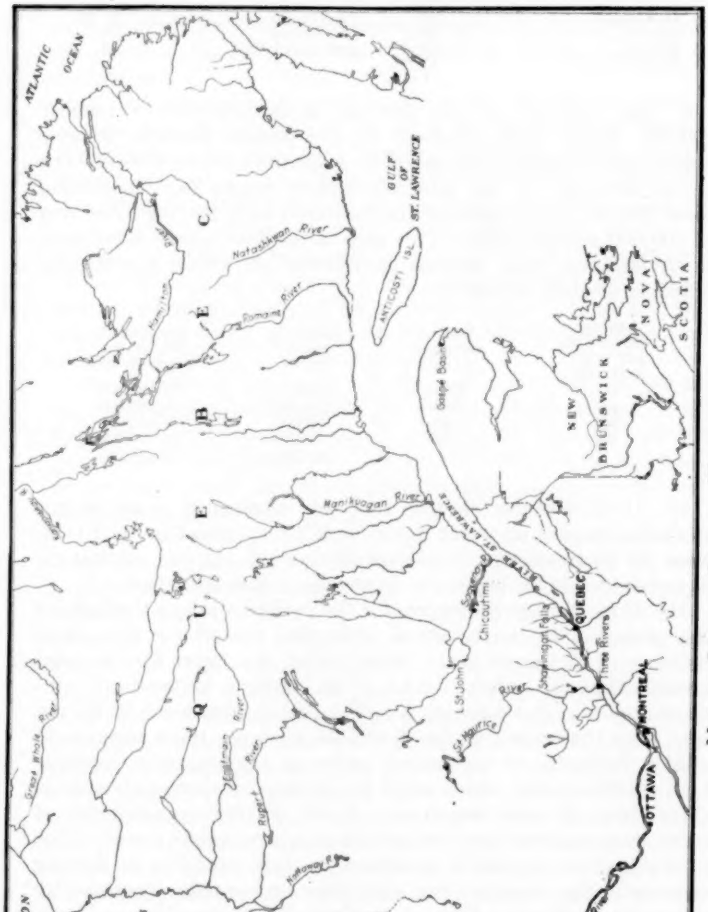
5 The north watershed is drained by many large streams flowing into James Bay, and when the development of the country has progressed far enough to permit their water powers to be utilized, enormous amounts of power will become available on the Nottaway, Rupert, East Main, and Kaniapiskau Rivers. With regard to this northern watershed, although the area is large the precipitation is largely unknown, and the run-off also is largely a matter of speculation.

6 The east watershed is drained principally by the Hamilton River, on which is situated a water power estimated at 700,000 hp. at Great Falls, some 350 miles from the mouth of the river and 200 miles north of the St. Lawrence River at the west end of Anticosti Island.

Potential Water Power of the Province of Quebec

7 The ultimate source of the water power of any country is the precipitation, and the total potential water power produced thereby may be estimated by calculating the potential energy of this precipitation in its descent from the highlands to the sea.

8 To calculate the total potential water power of the province of Quebec we may superimpose a map of the precipitation upon a topographical map of the province and then divide the province into areas having the same average elevation and approximately the same precipitation. An estimate of the potential energy due



MAP OF QUEBEC, SHOWING WATERSHEDS AND RIVERS

to precipitation can then easily be made. The precipitation in inches per year at different points of the province is as follows:

Montreal	40	Shawinigan	37.61
Quebec	40.5	Chicoutimi	30
Brome	34.3	Anticosti Island	36
Father Point	33.58		

9 The total area of the province is, as stated above, roughly 700,000 square miles, of which for the present purpose we may neglect the triangular area cut off by a line drawn from the northern tip of Labrador to the southern end of James Bay, containing some 200,000 square miles of snow-covered land, leaving a net area of 500,000 square miles. This area of 500,000 square miles may be divided into four plateaus of different elevations and slightly different rainfall, as follows:

Average recipitation per year in inches	Average elevation in feet	Area in square miles	Equivalent continuous power in kw.
40	1700	170,000	71,400,000
40	1000	220,000	92,400,000
35	500	100,000	18,350,000
40	300	10,000	1,260,000
		500,000	183,410,000

10 It is of course obvious that this amount of power is not available because no water would be left for vegetation, and even were all hydroelectric developments carried out, no allowances have been made for losses due to seepage and evaporation.

11 If we average the figures for the various stations throughout the province, we arrive at an estimated run-off for the whole province of 18 in. per year. Substituting this figure for the total precipitation, we obtain a total of 88.7 million kilowatts of continuous power, and allowing an efficiency of conversion of 60 per cent from the stream to the distribution center, there remains 53 million kilowatts of continuous power as the maximum possible hydroelectric power which could be produced if during all seasons every drop of water which ran off the 500,000 square miles of territory considered were converted into hydroelectric power. This is of course an impossible condition, as there would be no flowing streams in the country, but only lakes and canals connected by penstocks discharging through turbines into those of lower levels.

12 In estimating the amount of power which is actually available we must obviously neglect the energy in the water falling on the upper plateau, which is lost in the first part of its descent from these highlands. While it is impossible to state accurately the percentage of energy loss before the gathering together of the waters takes place, it may be conservatively estimated at 50 per cent on an average over the many steep and shallow valleys that go to make up the different drainage basins.

13 We must also subtract from the total of available energy all the energy consumed by friction of the banks and beds of the

rivers and streams, which is sufficient to use up about one-half foot of head per mile on the average; and in addition we must consider as non-available all the energy contained in those portions of the rivers and streams and brooks where the fall is hardly greater than that necessary to produce the required flow of water through the channel.

14 As continuity of supply is an essential condition to all power service, due allowance must be made for the fact that while the precipitation is more or less equally distributed during the year, the run-off from the precipitation is most unequally divided among the different seasons. During the spring thaw the melting and running away of the accumulated winter snowfall produces flood conditions on all rivers and streams, raising the flow to as much as 20 to 100 times the minimum. The necessary waste of the greater part of the water reduces still further the quantity of energy which may be considered as available for industrial uses, even under ultimate conditions of development.

15 Combining the sources of lost energy enumerated we may assume that not more than 25 per cent of the 53 millions of kilowatts of energy contained in the total run-off could be converted into electric energy, and therefore that the actual potential energy of the province of Quebec is approximately 13 to 15 millions of continuous kilowatts.

16 In the present state of the art of the development and distribution of hydroelectric power, only those sites near industrial centers and at which water power exists in considerable quantities are capable of being developed. The theoretical available quantity of 15 to 18 million kilowatts of continuous power must therefore be very considerably reduced in estimating the quantity of power that is actually adaptable to industrial purposes. From recent surveys of the water powers of the province, and considering only those rivers on which water power is available in such quantities and in such locations as could be economically developed either now or in the near future, the quantity of actually available water power in the province is estimated at approximately 5.25 million kilowatts of continuous power, of which some 800,000 kilowatts have now been developed and utilized. There remains, therefore, in the province of Quebec as available commercial water power about 4.5 million kilowatts of continuous power or about 12 million horsepower at 50 per cent load factor.

The Load Factor

17 The various amounts of power mentioned in this paper thus far have been described as "continuous power," which means that the flow of water producing this power is sufficient to provide the full quantity 24 hours a day, 365 days in the year. The author proposes to digress here from the main subject of the paper to discuss briefly the subject of "load factor," or the proportionate duration of use of power.

TABLE 1 PRINCIPAL WATER-POWER DEVELOPMENTS IN THE PROVINCE OF QUEBEC

River	Location	Company	Available Head, ft.	Total Generating Capacity, hp.
In Operation				
St. Maurice	Shawinigan Falls	Shawinigan Water & Power Co.	150	195,000
	Shawinigan Falls	Northern Aluminum Co.	145	40,000
	Grand Mere	Laurentide Power Co.	84	160,000
St. Lawrence	St. Timothee	Quebec-New England Hydraulic Corpn.	50	21,000
	Cedars Rapids		32	229,200
	Soulanges	Montreal Light, Heat & Power Co.	50	15,000
	Lachine Rapids		14	13,000
Richelieu	Chambly	Montreal Light, Heat & Power Co.	...	31,000
Ottawa	Hull	Ottawa and Hull Power & Mfg. Co.	40	20,000
		E. B. Eddy Co. (private)	...	10,000
St. Francis	Drummondville	Southern Canada Power Co.	32	6,700
Montmorency	Montmorency Falls	Quebec Ry., Light, Heat & Power Co.	270	5,000
St. Anne	Seven Falls	Laurentian Power Co.	410	20,000
Under Construction				
Saguenay	Lake St. John	Quebec Development Co.	300	400,000[1]
St. Maurice	Gres Falls	Shawinigan Water & Power Co.	65	132,000
Ottawa	Calumet Island	Ottawa and Hull Power & Mfg. Co.	60	60,000

[1] A further 800,000 hp. will be developed later near the mouth of the river.

18 No industry is able to make continuous use of the power which it requires, and therefore no power company supplying power for industrial purposes is called upon to supply the total amount of contracted power all day every day in the year. The term "load factor" is used as a measure of the duration of use of power and is defined as average power divided by peak power. The load factors of the various industrial loads in Quebec differ considerably. For instance, paper mills operate at a load factor of between 65 and 75 per cent, averaging more nearly 65 per cent; carbide and electrochemical plants operate at 85 to 90 per cent; cement mills at 80 to 90 per cent; and asbestos and mining industries at 45 per cent load factor. The load factor of a hydroelectric station supplying various industrial establishments and providing power and lighting service to communities will obviously be the average of the combined load factors of the various loads. Such a plant will therefore not require to use continuously the flow of its water which will produce maximum output. Provided sufficient water-storage capacity exists, the surplus energy that the normal flow of the river could produce may be stored in the form of water, but otherwise a certain portion of the available energy — which might be very great at certain seasons in the year — must be allowed to run to waste. In estimating the amount of energy available for industrial uses from the flow of the river at a given head, due consideration must therefore always be given to the storage capacity of the watershed and to the amount of regulation of flow which is possible with full utilization of the storage capacity.

19 In plants where no storage capacity exists, the installation of sufficient generating capacity to consume the increased flow of water during the six high-water months becomes economically justifiable by the use of electric steam generators as an adjunct to the plant to make possible the utilization of the excess energy produced during these months by its conversion into steam. The ability to obtain a return from all the energy available at all times of the year will make possible the development of many power sites which, if dependent solely upon the sale of continuous electric power as such, might remain undeveloped.

Early Water-Power Developments

20 The first hydroelectric development was the Montmorency plant near the mouth of the Montmorency River, seven miles below Quebec, built in 1895 and having an installed capacity of five 600-kw. generators.

21 This was followed in 1897 by the St. Narcisse plant on the Batiscan River, with an output of 750 kw., which was transmitted to Three Rivers over a two-phase 12,000-volt transmission line 18 miles long. This was the first high-tension line in the British Empire.

22 The Chambly plant on the Richelieu River, 17 miles from Montreal, was built in 1898 with four 2000-kva. machines, the

output of which was transmitted to Montreal at 25,000 volts, three-phase. In 1898, also, the Lachine Rapids plant began operation with a maximum output at certain times of the year of 10,000 kw.

Principal Water Powers Now in Operation

23 The principal water powers now in operation in the province of Quebec may be divided as follows: Powers on the Ottawa River, powers on the St. Lawrence River, Province of Quebec; powers on the St. Maurice River; powers on the Saguenay River and its tributaries in the vicinity of Chicoutimi; and powers located on the south side of the St. Lawrence River, particularly on the St. Francis River and its tributaries.

The two great developments existing today in this province are located at Cedars Rapids in the St. Lawrence River, thirty miles west of Montreal, and on the St. Maurice River at Shawinigan Falls and Grand Mere. The third great power, which ultimately may eclipse these, is now being constructed on the Saguenay River just below the outlet of Lake St. John.

24 In addition to the developments enumerated in Table 1, which generate electricity for use in public-utility service or for industrial service in the neighborhood of power developments, there are a large number supplying power for grinding wood. These latter are widely scattered throughout the province, most of them having been built prior to 1910. The total amount of hydraulic power used directly for this purpose in the province has been stated by the Water Power Branch to be 162,825 hp. The vertical turbine and the increase in the size of electrical generating units made it in many cases more economical to develop power in the form of electricity to be transmitted to pulp and paper mills built in favorable positions rather than to locate the mills at the site of the power development. The amount of hydroelectric power so used is stated to be 157,367 hp., making a total of 320,192 hp. of hydraulic turbines employed in the pulp and paper industry.

25 The growth of hydraulic and hydroelectric development has been rapid and continuous over the past 20 years. The yearly rate of increase has been about seven per cent of the installed capacity per year over the last 15 years, and the present total development is 1,070,000 hp. of hydraulic and electrical capacity combined.

26 At the present rate of increase the total available amount of hydraulic power, which is estimated at 5.25 million kilowatts or about 7,000,000 hp., will be developed and utilized in the next 29 years. However, as the increase in development will in general follow the increase in population, it will take considerably less than 29 years to develop all the power now in sight, and a careful study indicates that in 20 to 25 years the 7,000,000 hp. will be used up. Further, this time might be materially shortened if any large amount of power were exported from Canada.

No. 1887

HYDROELECTRIC DEVELOPMENTS IN ONTARIO[1]

By F. A. Gaby,[2] Toronto, Canada
Member of the Society

Preliminary estimates of the available water power in Ontario indicate a total of 6,000,000 hp., of which 1,300,000 hp. have already been developed. The early developments at Niagara Falls are reviewed, and a detailed description of the recent Queenston-Chippawa development is given. This development, which is the largest in the world, will ultimately develop 650,000 hp. in ten units operating under 305 ft. head. When this development is complete, the generating capacity at Niagara Falls will be over 900,000 hp. An outline of the work of the Hydro-Electric Power Commission of Ontario is given, showing how it has surveyed and mapped out available powers, years in advance of the necessity of utilizing them, and planning their development, so that when the market for power had developed the power was available.

THE province of Ontario is richly endowed with water powers, widely distributed over an area of over 400,000 square miles, and its surface waters drain both to the Atlantic and to the Arctic oceans. Preliminary estimates which include Ontario's share in the water powers of her international waters have indicated a total of some 6,000,000 hp., of which about 1,300,000 hp. has already been developed.

The Chief Water Powers of Ontario and Their Strategic Situation

2 The power potentialities of the Niagara and St. Lawrence Rivers constitute a very large proportion of the available power in the province. Next in importance is the Ottawa River and its tributaries. The power on the main stream, which constitutes part of an interprovincial boundary, is shared by the provinces of Ontario and Quebec. Under conditions of controlled flow,

[1] For discussion, see p. 36.
[2] Chief Engineer, Hydro-Electric Power Commission of Ontario.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers. Abridged.

Ontario's share of the power available on the main stream, together with that on the tributaries in Ontario, aggregates about 700,000 hp.

3 Other important water powers are found in the Trent River watershed, tributary to Lake Ontario. There are nearly thirty sites, of which about one-half have been developed with an aggregate capacity of 50,000 hp., leaving undeveloped sites with an aggregate capacity of some 15,000 hp. Tributary to Lake Huron are water-power streams with an aggregate potentiality at known sites of nearly 300,000 hp., of which about 100,000 hp. is developed. Tributary to Lake Superior are water-power streams with an aggregate potentiality of about 300,000 continuous hp. About half of this total is on the Nipigon River, on which an initial development has been made for the municipalities of Port Arthur and Fort William by the Hydro-Electric Power Commission of Ontario. The outflow from Lake Superior through the St. Mary River constitutes an important water-power site, which has been partially developed. In the extreme west of the province, the English and Winnipeg Rivers, which flow into the province of Manitoba, have important water powers; in Ontario these powers aggregate over 250,000 hp. There remain to be mentioned the water powers of the streams flowing from Ontario into James Bay and Hudson Bay. The more important of these streams from the viewpoint of water-power possibilities are the Mattagami, the Abitibi, the Missinaibi, and the Albany Rivers. These streams have not yet been adequately appraised; the total water-power possibilities of known sites aggregate upward of 1,000,000 hp.

Early Hydroelectric Developments in Ontario

4 The most important *early* hydroelectric development in the province of Ontario, from the standpoint of magnitude and extensive transmission, was the plant at DeCew Falls. This plant takes its water from Lake Erie at the level of the upper reach of the Welland Canal at Allenburg. It is a high-head development (265 ft.) of about 50,000 hp. capacity. Current is generated at 2400 volts, 3-phase, 66 cycles, and transmitted at 44,000 volts. The main objective of this DeCew power was the city of Hamilton, in the industrial life of which it has played a most important part.

5 *Niagara Falls.* The three large hydroelectric plants at Niagara on the Canadian side of the river have often been described in print, but a brief reference should be made to these developments because of their importance to the general hydroelectric economy of the province of Ontario.

6 The Canadian Niagara Power Company — an ally of the Niagara Falls Power Company of Niagara Falls, New York, was formed in 1889. The water for this plant is drawn from the level of the upper river through an intake canal, and is thence distributed to the intake chambers at the head of each penstock. The penstocks pass vertically down an average depth of about 160 ft. to

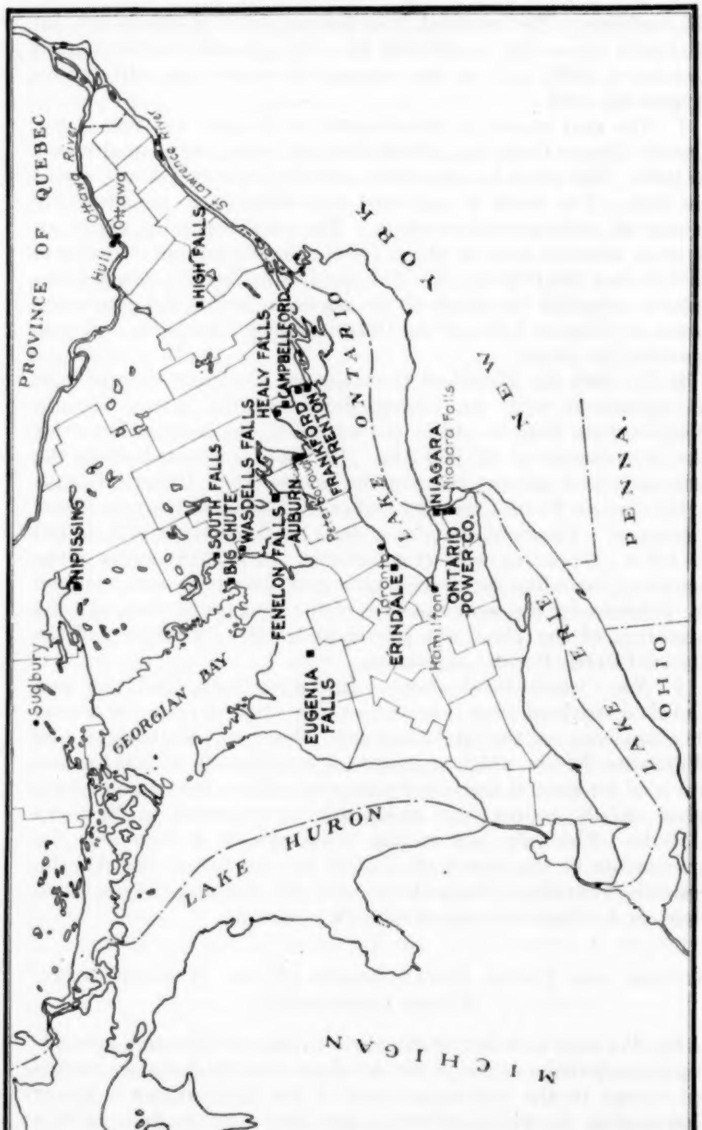


Fig. 1 Map of Ontario, Showing Watersheds and Rivers

the turbines. The nominal installed capacity of this plant, including a spare unit, is 121,000 hp. The ground was broken on October 4, 1890, and the first commercial power was delivered on August 26, 1895.

7 The next important development at Niagara was that of the Ontario Power Company, which first delivered commercial power in 1905. This plant has its intake near the Dufferin Islands, above the falls. The water is conveyed over 6000 ft. to penstocks by means of underground conduits. The plant operates under an average effective head of about 180 ft., and its present capacity is a little over 160,000 hp. In 1917 the Hydro-Electric Power Commission acquired the whole of the assets, including the generating plant at Niagara Falls, of the Ontario Power Company, and now operates the plant.

8 In 1903 the Electrical Development Company entered into an agreement with the Commissioners of the Queen Victoria Niagara Falls Park to utilize the waters of the Niagara River for the development of 125,000 e.hp. Its plant is situated above the falls and about midway between the headworks at Dufferin Islands of the Ontario Power Company and of the Canadian Niagara Power Company. The plant operates under a head which varies from 130 to 145 ft., depending on river conditions. In 1922 the assets of the company, including the development just described, were acquired by purchase by the municipalities of the Niagara district, and the operation of the plant was placed under the jurisdiction of the Hydro-Electric Power Commission.

9 *New Ontario Developments.* In the Sudbury, Cochrane, and Cobalt districts of what is known as New Ontario there are several developments on the Matabitchuan, Montreal, Mattagami, and Wanapitei Rivers which aggregate approximately 60,000 hp. and are used for general municipal purposes and the operation of gold, silver, cobalt, copper, and nickel mining properties in Northern Ontario. The pulp and mining companies have their own developments to the extent of 125,000 hp. located on the Abitibi, Spanish, Vermillion, Magnetawan, and the Onaping Rivers, which they use for their own manufacturing purposes.

Systems and Power Developments of the Hydro-Electric Power Commission

10 We may now devote attention to some of the more interesting characteristics of the power developments which supply electrical energy to the various systems of the Hydro-Electric Power Commission to which reference has just been made. In this connection special attention will be directed to certain features of some of the developments installed under the administration of the Commission.

11 From an initial investment in 1910 of some $3,750,000 the operations of the Commission have extended until at the present

time it administers properties to the value of approximately $200,000,000, delivering upward of 625,000 hp. to 14 different systems in various parts of Ontario and serving 350-odd municipalities, the systems comprising some 3500 miles of transmission lines.

12 *Transmission Voltages.* The frequency employed is 25 cycles in the Niagara system and 60 cycles in all other systems. In the Niagara system the main high-tension lines are operated at 110,000 volts and are carried on steel towers. The secondary distribution from the main transformer stations to the various municipal and distribution stations is chiefly at 13,200 and 26,000 volts, although other voltages are employed. Outside of the Niagara system the only 110,000-volt lines are in the Thunder Bay district, where about 70 miles of wood-pole line are operated at this voltage. The main transmission lines of the Severn, Eugenia, Wasdells, and Muskoka systems are operated at 22,000 volts; the St. Lawrence system and the Rideau system lines at 26,400 volts; the Central Ontario and Trent system lines at 44,000 volts — recently raised from 26,400 volts, at which some lines still operate — and the Nipissing system lines at 22,000 volts. Local distribution in the various systems is usually at 2200 and 4000 volts, the service voltages being 110, 220, and 550.

13 The extent of the hydroelectric developments by the Commission, both actual and conjectural, are summarized in Table 1.

14 *Storage.* The rapidly increasing demand for electrical energy in all of the fourteen systems operated by the Commission has in some instances already used the available water supply, while in other systems it is only a matter of two or three years, or even less, before practically all of the power available under existing conditions of water supply will be requisitioned. The Commission will be able in some cases to increase the supplies available by providing storage. Most of the drainage basins supplying existing developments are well furnished with lakes and reservoir possibilities. The potentiality of the storage available, as well as the potentialities of remaining power sites, are being investigated by the Commission's engineers, it being recognized that such storage as may economically be developed will be demanded at an early date.

The Queenston-Chippawa Development

15 The largest development in Ontario, in fact, the largest single hydroelectric development in the world, is the Queenston-Chippawa installation. This development was placed in service on February 11, 1922. It will have a normal operating head of 294 ft. to 305 ft. when the installation is complete, which is over 90 per cent of the fall between Lake Erie and Lake Ontario. The conservation of head effected by the reduction of hydraulic losses to a minimum and by refinements in the design of the various essential elements of the project as a whole, has resulted in the production

TABLE 1 CAPACITY OF PRESENT AND PROJECTED DEVELOPMENTS OF THE HYDRO-ELECTRIC POWER COMMISSION OF ONTARIO

Present Development or Site	Present Plants: Present capacity, hp.	Present Plants: Ultimate capacity, hp.	Projected Developments: Estimated hp.
Niagara System			
Niagara River:			
Ontario Power Co.	180,000	150,000	......
Electrical development Co.	125,000	100,000	......
Queenston-Chippawa development	300,000	550,000	......
Canadian Niagara Power Co.[1]	20,000	20,000	......
Total	625,000	820,000	......
Combined Northern Systems — Severn, Eugenia, and Wasdells			
Beaver River:			
Eugenia Falls development	8,500	8,500	......
Severn River:			
Wasdells Falls development	1,200	1,200	......
Big Chute development	6,200	6,200	......
Port Severn	......	......	1,400
Saugeen River:			
Haywards Falls	......	......	1,020
Port Elgin	......	......	10,000
Kimberley	......	......	1,700
Total	15,900	15,900	14,120
Muskoka System			
Muskoka River:			
South Falls development	1,750	6,000	2,000
St. Lawrence System			
St. Lawrence River:			
Cedars Rapids Power Company[1]	......	......	......
Morrisburg and Long Sault	......	......	600,000
Rideau System			
Mississippi River:			
High Falls development	3,600	Fully developed	......
Carleton Place development	735	Fully developed	......
Ragged Chutes	......	......	3,600
Total	4,335	......	3,600
Thunder Bay System			
Nipigon River:			
Cameron Falls development	25,000	75,000	......
Alexander Landing	......	......	41,900
Pine Portage	......	......	40,400
Virgin Falls	......	......	30,800
Kaministikwia River:			
Silver Falls (Dog Lake)	......	......	22,000
Total	25,000	75,000	135,100
Ottawa System			
Ottawa River:			
Ottawa and Hull Power & Mfg. Co.[1]	......	20,000	......
Chats Falls	......	......	60,000
Total	......	20,000	60,000
Central Ontario and Trent System			
Developments above Rice Lake	11,140	......	......
Developments below Rice Lake	40,470	......	10,720
Total	51,610		10,720

[1] Purchased power.

NIPISSING SYSTEM

South River:			
Nipissing development	2,500	Fully developed	......
Bingham's Chute	......	......	1,300
Elliotts	......	......	1,150
Gitzlers	......	......	2,160
Cox's Chutes	......	......	1,930
Gimballs	......	......	700
French River: (three developments)			
Chaudiere, Five Mile Rapid, and the Dalles	......	......	30,000
Total	2,500		37,240
GRAND TOTAL..............1,858,125 hp.			

of a power development which is believed to represent the best in modern engineering practice.

16 The plant when completed will consist of ten 60,000- to 65,000-hp. units running at 187.5 r.p.m. and generating power at 12 kv., three-phase, 25 cycles, which in turn is transformed to 110 kv. for transmission with an ultimate capacity of 550,000 to 600,000 hp.

17 Fig. 2 indicates the relation of the various works comprising the development. Water is taken from the Niagara River about one mile above the falls, is conveyed $4\frac{1}{2}$ miles through the improved section of the Welland River, thence by a canal $8\frac{1}{2}$ miles long to the forebay and screen house located on the Niagara River about one mile south of the village of Queenston. From the screen house, steel penstocks encased in concrete carry the water down the cliff to the power house, from which it passes to the Niagara River.

18 *The Intake.* On the Niagara River one of the great obstacles to securing continuity of service is the annual formation and flow of ice. In order to eliminate as far as possible interference of operation due to ice, a special form of intake has been provided.

19 The complete structure is approximately 1100 ft. long and is made up of an entrance with lock gates for navigation, a bulkhead section, and the intake proper, the latter combining two forms of intake; the conventional or surface intake consisting of a concrete barrier or boom with fifteen openings, each 18 ft. in width, normally having 8 ft. of submergence, which submergence, however, can be increased by means of drop gates to any amount up to the full depth of water or 35 ft.; and the submerged intake consisting of gathering tubes or draft distributors, six in number and 675 ft. in length.

20 *Control Gate.* The control gate, located at the upper end of the canal, where the earth section merges into the rock section, is the largest single-leaf, roller-type, motor-operated gate ever built. A worm drive is connected to double hoisting gears on the top of the end towers. Roller chains pass over these gears, connected at one end to counterweights and at the other to the gate. When raised to its full height it leaves a clearance of 14 ft.

above the water level in the canal. The total pull on the two chains in raising the gate is 316,000 lb. The chains are made up of pairs of links 9 in. by $1\frac{1}{4}$ in. in cross-section, and rollers $4\frac{1}{2}$ in. in diameter. The speed of lift is 4.6 ft. per min.

21 *Concrete Lining.* Economic considerations prompted the lining of the canal with concrete averaging 20 in. thick. The

FIG. 2 THE QUEENSTON-CHIPPAWA DEVELOPMENT

height of the lining was fixed slightly lower than the profile of the water surface existing when the load conditions on the plant are a maximum and the Niagara River flow is a minimum. The lining thus always is protected by submergence against frost.

22 *Screen House.* At the lower end of the forebay, and serving as a dam for it, is the screen house, which forms the entrance and the control for the penstocks. The entrance to each of the main penstocks is a modified bell mouth consisting of three openings, 12 ft. 8 in. wide and 29 ft. high, at the rack supports, which gradu-

ally converge into one opening 16 ft. in diameter at the point of connection to the penstock. Immediately behind the curtain wall steel-lined gate checks are provided to support structural-steel gates.

23 On account of the provision of Johnson valves at the lower end of the penstocks, permanent gates were not installed in the screen house at the entrances to the penstocks. Movable gates have been provided, consisting of sectionalized leaves which can be dropped into checks in the piers by means of the 25-ton traveling crane in the screen house. Concrete piers divide the entrances to the penstock into three openings, and the drop gates for closing these openings are made in three sections.

24 *Penstocks.* From the screen house the water is carried to the turbines in steel-plate penstocks encased in a concrete envelope with a minimum thickness of 24 in. The economical diameter for units 1 to 5 was determined to be 15 ft., and in order to simplify the field riveting the lower third of the penstock was made 14 ft. and the upper two-thirds 16 ft. This decrease in the diameter at the lower end permitted the use of plates 1¼ in. thick, which could be readily riveted in the field. The total weight of each penstock is 840,000 lb.

25 Each penstock ring is made up of two plates with longitudinal joints on the horizontal center line, the plates varying from ½ in. at the top section to 1¼ in. in thickness at the lower section. These joints are all double-strap butt joints, varying from double riveted at the top to quadruple riveted at the lower end. The circumferential joints are single-strap butt joints, double riveted, with the butt strap on the outside. The longitudinal joints are calked on the inside, but the circumferential joints are made watertight by electric welding. In designing the penstocks a stress of 12,000 lb. per sq. in. was used, this figure being taken to provide for the exigencies of corrosion, fatigue, suddenly applied loads, and other indeterminate or unknown contingencies. The internal pressure used for design purposes was taken to be the static head plus the pressure rise due to a complete closing of the turbine gates in 1½ sec. This increase in pressure was taken as a maximum at the turbine gates and varying uniformly to zero at the racks.

26 For discharging ice which may form on the canal and forebay, a chute has been provided at the south end of the screen house leading down the cliff and under the power house to the Niagara River. This ice chute is 10 ft. in diameter, made of reinforced concrete and provided at the upper end with a drop gate.

27 *Generating and Transformer Station.* The generating and transformer station is located below the escarpment and close to the river's edge. The station extends about one-half the distance to the top of the escarpment. The structure required to house five main units and the service equipment is 350 ft. long, and ultimately this length will be doubled. The substructure is of massive concrete construction carried down to rock foundations, and provides

chambers and tunnels for housing and giving access to various kinds of apparatus. The superstructure consists of a structural-steel framework with reinforced-concrete floors and roofs, and concrete, brick, and tile walls and partitions.

28 *Turbines and Governors.* Turbines Nos. 1 and 2 are designed for 52,000 hp. and Nos. 3, 4, and 5 for 55,000 hp., all at 305 ft. head, the speed being 187.5 r.p.m. They are of the single-runner vertical type, set in spiral cast-steel casings. The inlet diameter of the casings is 10 ft. In Nos. 1 and 2, manufactured by the Wellman-Seaver-Morgan Company, of Cleveland, the casings are divided into nine sections with a separate speed ring, while in Nos. 3, 4, and 5, manufactured by the Wm. Cramp & Sons Ship & Engine Building Company, of Philadelphia, the speed ring is cast integral with the casing and the whole divided into 12 sections. The weight of the casing and speed ring in Nos. 1 and 2 is 240,000 lb., while in 3, 4, and 5 it is 180,000 lb.

29 The runners are made of cast steel in one piece and provided with special renewable seal rings. Those for Nos. 1 and 2 are 125 in. in diameter, and those for Nos. 3, 4, and 5 are 121 in. in diameter.

30 The gate control is the usual double-regulating-cylinder type, operating through rods to a shifting ring connecting to the individual gates through specially designed breaking links.

31 All parts of the turbines adjacent to rotating elements and to water passages subjected to high velocities are made of special steel and renewable. The main-journal guide bearing on each unit is of the water-lubricated lignum vitæ type, such as has been used on all modern vertical-turbine plants. This type of bearing has been selected on account of its simplicity and low maintenance cost.

32 The draft tubes are all of the Moody spreading type except that of No. 1, which is a bent tube.

33 On account of the limitations in the use of water at Niagara, the runners are designed to obtain their maximum efficiency at a point about 10 per cent below their maximum rated output, at which load for the most part they are operated. This condition required that the runners be larger than would normally be used and that the turbines be "gated back." With the gates full open, actual tests have shown that the capacity of Nos. 1 and 2 is 60,500 hp. each, and of Nos. 3, 4, and 5, 65,500 hp. each, under 305 ft. head.

34 Exhaustive tests have been carried out on the units, using the Gibson "pressure-time" method for determining the discharge. The maximum efficiency obtained on each turbine was 93 per cent, exceeding, so far as is known, any figure heretofore recorded on a commercial operating turbine. Fig. 3 shows a turbine casing erected in place.

35 The governor actuator is set on the generator floor, the flyballs being driven by belt from the turbine shaft. The main valve is located adjacent to the turbine regulating cylinders, 35 ft. below the actuator. To meet the conditions of this layout the pilot

valve is hydraulically coupled to a small piston valve which is mechanically connected to the pilot valve of the main dog valve. The hand-control stand is located on the generator floor and operates the regulating cylinders through the governor pressure system.

36 The governors are operated on the "central" type of pressure system. Two motor-driven centrifugal pumps, each delivering 500 imp. gal. per min., and located adjacent to two 3000-gal. sump tanks built into the concrete substructure of the power house, deliver the governor fluid at a pressure of 200 lb. per sq. in. to a header from which branches are connected to the governors

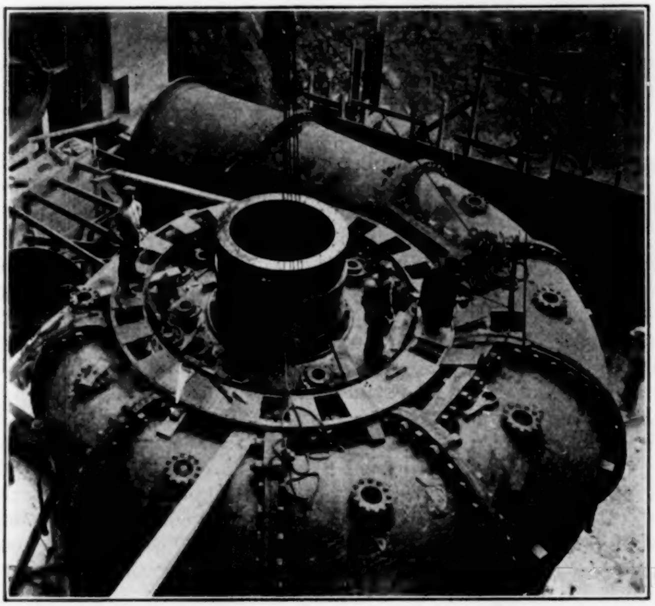


Fig. 3 Turbine Casing Erected in Place

through individual pressure tanks. The capacity of one pump is sufficient to operate five governors.

37 The fluid used in the governor system is filtered water, in which is dissolved 7 lb. potassium bichromate to every 1000 gal. water. This fluid does not corrode the valve seats and has proved satisfactory in service.

38 *Service Turbines.* The two service-turbine casings have a 30-in. inlet diameter and runners of 42 in. outside diameter. The gate mechanism is of the "outside" type, similar to the main turbine.

39 The governor is the "direct-connected" type, and the pres-

sure liquid is ordinary lubricating oil. The flyballs are mounted upon the main turbine shaft, and the governor stand, with self-contained pumping unit, is placed on the turbine floor.

40 A band brake is furnished in order to assist in bringing the unit to a quick stop when necessary.

41 *Johnson Valves.* A Johnson valve is located on each penstock near the entrance to the turbine casing. The action of this type of valve has been described frequently in other papers. The control of the Queenston valve is made through the operation of three small Johnson valves which automatically adjust the flow into the central and annular chambers so that the main valve plunger cannot travel at a dangerous rate. The closing stroke is cushioned at the end, and in the opening stroke the plunger moves just sufficiently to permit the wheel case to be primed before completing its stroke. An emergency handle is located on the control stand on the generator floor, by means of which an operator can close the Johnson valve in case of trouble. The valve can be opened only from the control located adjacent to the valve itself.

42 *Acres Control Pedestal.* For the purpose of the floor operation of each unit, a pedestal is erected adjacent to each generator. On this pedestal are mounted the pressure and vacuum gages for the turbine, signal light, and loud-talking telephone from the control room, temperature indicators from the generator windings, oil- and water-flow indicators, a control for the air brakes, and the emergency closing control for the Johnson valve. An operator standing at the pedestal is in close communication with the control room and at the same time has under his hand the control of the turbine governor and Johnson valve. The signal lights operated from the control room call the floor operator to any unit desired.

43 *Generators.* The present five units are each rated at 45,000 kva., 80 per cent power factor, 12,000 volts, three-phase, 25 cycles at 187.5 r.p.m. They are capable of being operated continuously at 49,500 kva. with either voltage or current 10 per cent in excess of the rated values. The type is vertical (see Fig. 4), with direct-connected, shunt-field, commutating-pole, 250-volt, 150-kw. exciter. The overall efficiency of the generating units is slightly in excess of 97 per cent at 80 per cent power factor. The thrust bearing is designed to support a load of one million pounds, which is slightly more than the weight of the rotor plus the hydraulic thrust imposed by the turbine. Upper and lower guide bearings are provided, the latter on account of the length of shaft and to keep the generator a self-contained unit.

44 The overall diameter of these units is 25 ft., the diameter of rotor over pole faces being 18 ft. approximately. The shafts are 30 in. in diameter in the guide bearings, and are provided with a flange at the lower end for bolting to a corresponding one on the turbine shaft. The shafts are hollow, with an 8-in.-diameter bore, and are 30 ft. 3 in. in total length. The overall height of the generators above the generator floor is 26 ft. 10 in., thus above the

main floor only the top of the frame and the upper bracket, thrust-bearing housing, and exciter are visible (Fig. 4).

45 The weight of the complete generator is 1,400,000 lb. and that of the rotor, 615,000 lb. The largest piece to be handled by the cranes weighs 600,000 lb.

46 It was specified that there should be a thrust bearing at the

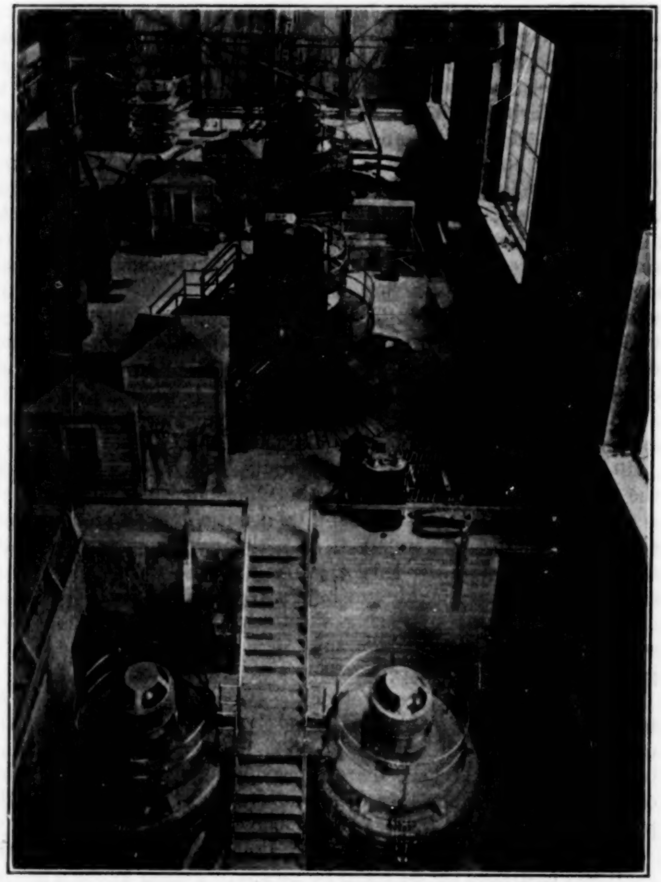


FIG. 4 INTERIOR VIEW OF GENERATOR ROOM

top to carry the weight of the rotating part and the total thrust due to turbine at rated speed and at overspeed; that the rotor be capable of operating at an overspeed of 347 r.p.m. without injury to any part; also that provision for easy handling of various parts be made.

47 Contracts were placed with two manufacturers, who used

different methods in constructing the generators, the general difference being that one made use of a built-up laminated sheet-steel rotor rim mounted on a small cast-steel spider, of cast iron in the upper and lower bearing brackets, and of a Kingsbury thrust bearing to carry the rotating parts; while the other used a rotor consisting of seven cast-steel wheels shrunk on to the shaft, cast steel in the upper and lower bearing brackets, and a thrust bearing with stationary bearing plate supported on a large number of small springs. Both manufacturers used forged-steel hollow shafts of approximately the same dimensions, which were 30 in. in diameter in the guide bearings and had a forged half-coupling at the lower end.

48 The steel forgings and steel and iron castings were required to meet the specifications of the American Society for Testing Materials. With reference to the steel castings, particularly those of large section, such as are used in the cast-steel rotor wheels, attention is drawn to the advisability of not depending entirely on small test pieces attached to the casting to determine if the annealing has been properly carried out. It is the practice of the Commission on these large rotor castings to require that test pieces taken from 4-in. by 4-in. blocks 12 in. long, cast on the outside and underside of the rim, meet the specification requirements, also that core-drill samples be taken from the rim at the ends of the arms and if possible from between the arms, these samples being tested and examined to determine the quality of the metal.

49 The two types of bearings provided are interchangeable in the housings as constructed. In the Kingsbury bearing the shaft is supported on the thrust bearing by a nut threaded on the upper end of the shaft. The nut rests on a thrust collar which rests on the runner plate of the thrust bearing. The runner plate is of cast iron with a very highly finished surface, being true to 0.0002 in.

50 The stationary bearing surface consists of six babbitted sector-shaped shoes, each mounted on a support on the head of a jackscrew so that it is free to tilt.

51 The maximum diameter of the runner plate is 69 in. The total area of the shoes is approximately 2500 sq. in., so that the unit pressure is 380 lb. per sq. in. The friction losses in this bearing at normal load and speed are approximately 90 hp. Lubricating oil is circulated at the rate of about 4 gal. per min. between the purifying system and the bearing housing.

52 In the spring type of bearing the thrust is taken from the shaft to a cast-steel thrust collar by means of a shear key in a groove in the shaft. The groove is 3 in. wide and the shaft 29 in. in diameter. The thrust collar rests on a runner plate of cast iron 69 in. in diameter and is attached by two 2-in. dowel pins to prevent relative rotation. The carefully finished surface of the runner plate runs on a stationary steel ring with a babbitted surface, which in turn is supported on a nest of over 700 steel helical springs. The babbitted ring is split in one place radially to give flexibility, and the surface is radially grooved to allow oil to reach

the rubbing surfaces. The unit pressure is about 265 lb. per sq. in. of bearing surface at 960,000 lb. total downward thrust. The total loss in the bearing is about 125 hp.

Growth in Demand for Hydroelectric Power

53 The first five units of the Queenston-Chippawa development are now in operation and the other units are being installed as rapidly as possible, and when completed the Commission will have a generating capacity for the Niagara District of over 900,000 hp., including the Toronto Power Co. and the Ontario Power plants.

54 Since 1910 the demand for "Hydro" service from the Commission has grown from a small initial load of less than 1000 hp. to one of 628,000 hp. (of which 75,000 hp. is exported) taken by 250 municipalities and 97 townships supplying 335,000 consumers.

55 Realizing as it does the large amount of time — often several years — that must elapse between the initiation of a new, extensive water-power development and its being brought to the stage where commercial power can be delivered, the Hydro-Electric Power Commission has always been compelled to take early action on behalf of municipalities for the creation of new sources of supply for electrical energy. It has sometimes required a good deal of courage on the part of the officers of the Commission to incur heavy financial outlay years in advance of the actual demonstration of markets to absorb electrical output, but in no instance has its judgment eventually been found to have been in error.

56 At the present time, all evidence points to the fact that by the end of the year 1926 the existing provisions made for the supply of electrical energy to municipalities will be fully taxed and no spare power will be available. The output of the Queenston-Chippawa plant will have been absorbed. The Commission has considered the possibility of producing further power at Niagara Falls, but the chairman of the Commission, Sir Adam Beck, has stated from the public platform that he believes that it would be more profitable for all concerned to turn to the St. Lawrence River for the next large hydroelectric development.

Power on the St. Lawrence River

57 The province of Ontario shares with the state of New York the water power in the international stretch of the St. Lawrence River extending from Prescott to the international boundary. In this portion of the river there is a possibility of developing upon a conservative basis about 1,600,000 hp., of which 800,000 hp. would belong to Ontario. For several years the Commission has been conducting special surveys and investigations with the view of determining the best means of developing the international reach of the river in the joint interests of power and of navigation.

Rates Charged for Electrical Energy

58 One of the basic principles upon which the Hydro-Electric Power Commission of Ontario operates is the furnishing of electrical energy to consumers *at cost*. Time and again, it has been stated that the rates for light and power resulting from the efforts of the Commission were much less favorable than those prevailing in other places. However, the author wishes to state that nowhere in the world over such extensive areas do communities and citizens obtain electric power and light at such low rates as prevail throughout the hydro municipalities in the province of Ontario, namely, from 1.5 to 3.1 cents per kw-hr. for residence and commercial service, and from $13.26 to $25.14 per hp-year for power service.

59 The secret of the phenomenal growth of the operations of the Commission lies in the fact that as the trustee and agent of coöperating municipalities it has made available to them electrical energy *at cost*, which they have then distributed and vended to their individual consumers at *as near cost as possible*. Moreover, the Commission has made its distribution of energy to small municipalities and to rural districts, and the demand of these small municipalities aggregates a substantial load. Electrical energy, once a luxury, has now become a common commodity of the people.

DISCUSSION[1]

F. Darlington.[2] Economic operations are rapidly extending the application of power, and it is most important that the supply keep pace with the need. If it should fall behind, then in such proportion will prosperity be curtailed and standards of living restricted. Certain features of the power industry are international in their scope and of importance to all individuals. One of these is the increased centralization and unification of power service by interconnecting and expanding existing electric systems, which will be the greatest forward step that can be taken to put cheap and dependable power within the reach of all. It is not enough to have a huge power plant at Niagara Falls or on the St. Lawrence River or in the Pennsylvania coal fields. While power confined to these places might be used in factories or electrochemical works located near the source, its more general use and broader application must be through wide distribution to industrial and social needs. Only by wide distribution from the best natural power sources can the public be properly served.

Of the two basic resources for power generation — fuel and water — one is conserved by saving it, the other is lost if it is not used, but where both are available the best conservation is achieved by using them to supplement each other. Furthermore, whichever source of power is used there must be great generating

[1] Joint discussion of Papers Nos. 1886 and 1887.
[2] Consulting Engineer, New York, N. Y.

plants and power-transmission systems, and conservation requires that the fullest possible use should be made of these structures.

All the conditions for conservation and high economy are enhanced by interconnections between electric systems to combine the loads into big blocks and permit the use at all times of the most economical available source of power. Then when new generating plants are built, the most efficient source of power within the area of the interconnected system always can be used, whether it is a water power or a steam plant at a coal mine, or at tidewater or elsewhere.

It is sometimes erroneously held that if any given electric company has the cheapest source of power within practical transmission distance, such a company would have little or nothing to gain by tying in with other less fortunately situated concerns. This is radically wrong, even from a purely selfish point of view, for the more economical a generating plant may be, the more important it is that the fullest possible use should be made of it, and interconnection in great superpower systems brings opportunities for the fullest use of the most efficient plants with corresponding economy and conservation. For example, a superior coal mine plant, say, in Pennsylvania, will gain much by interconnection with a New England, New York, and New Jersey superpower system, or still more by interconnection with St. Lawrence water powers, for by interconnection it will secure a fuller use of its plant and a relay or breakdown connection for use in emergencies.

So to meet the certain need for more and more power, to enable industry to produce in greater fullness and society to live better, we should lay our plans on broader lines than heretofore and enter into relations that will conserve and serve by coördinated power systems regardless of private interest or state or national boundaries.

John R. Freeman. In many ways the Canadian people have shown more foresight in the development of their natural resources than has been shown in the United States. It has been the duty of the Water Power Branch of the Canadian government to study the available water powers in advance, and to see that each one was developed to the best of its possibilities. All possible sites were mapped long before they had real commercial value, and surveys were on record, so that when a development was made there was no tendency to use the best and leave the remainder. The work of the Water Power Branch has been a model to the whole world, and there has been too little appreciation of the magnificent work it has done for twenty years, not only in the surveying of water-power sites but also in the actual building of storage reservoirs.

Fifty years ago there were about 70 turbines in Lawrence, Mass., with a total of about 10,000 to 15,000 hp., and the Merrimac River was called the hardest-worked river in the world. At the present time more power is concentrated in one of the Chippawa

turbines than in all the 70 turbines at Lawrence, plus all the turbines then at Lowell and Manchester.

W. Monroe White. It has been said that the reason why the American manufacturer could compete successfully in the markets of the world and yet pay much higher wages than were paid to workmen of other countries was because each American workman had behind him twice the horsepower per man that his competitor's workmen had. Since the proportion of cost is in direct proportion to the amount of power expended in production, that workman who can direct the greatest amount of power will receive the greatest wage. Power means prosperity. Engineers are urged to recommend the development of power in any form.

P. S. Gregory.[1] While steam generators are a useful and even a necessary adjunct to most hydroelectric plants, the electric steam generators referred to are not to be confused with a coal-fired steam-generating plant. It is not the custom to have steam stand-by plants for hydroelectric stations in this country.

In all rivers in Canada the flow varies considerably during the different seasons, and there are times in the year when there is more water than the amount necessary to generate the normal load of the power companies, and as a consequence water must be spilled over the dams. Electric steam generators have been designed to utilize the energy of this surplus water, and their design has been carried to a high state of development. There are two electric steam generators in the plant of the Laurentide Company at Grand Mere which, united, have a capacity of 60,000 kw., or 6000 boiler hp., and in the mill at Shawinigan there is a steam generator of about 30,000 kw. capacity, or 3000 boiler hp. The power used in these steam generators is surplus power which otherwise would spill over the dam.

The return from the power used in electric steam generators may be calculated as follows: One kilowatt-hour of electrical energy will produce about three pounds of steam at ordinary pressure, that is, at, say, 125 lb. per sq. in., because there are roughly 1000 B.t.u. in one pound of steam under such conditions, and the energy in one kilowatt-hour is equivalent to 3412 B.t.u. At an average evaporation of 7½ lb. of steam per pound of coal, which is a reasonable figure, one ton of coal will produce 15,000 lb. of steam, and the equivalent electrical energy is therefore 5000 kw-hr. If a ton of coal costs $10, 5000 kw-hr. must be sold for $10 if the power consumed in the electric boiler is sold on a fuel-substitution basis. $10 for 5000 kw-hr. is two mills per kilowatt-hour. There are 6500 kw-hr. in one horsepower-year at 100 per cent load factor. At two mills per kilowatt-hour, each horsepower-year will bring in

[1] Asst. to Vice-President, Shawinigan Water and Power Co., Montreal, Canada.

a return of $13. For different costs of coal the return per horsepower-year is in direct proportion.

The interconnection which is made use of in Canada is interconnection of water-power plants on different rivers, on different watersheds where the periods of high water vary. For instance, interconnection of the plants which are now developed or will be developed on the Saguenay River with the plants developed on the St. Lawrence would give a far shorter period of low water and a longer period of high water because the high-water period on the Saguenay is from three weeks to a month or more behind the high-water period on the St. Lawrence. This would also apply to plants further west which have their high-water period earlier than on the St. Lawrence.

F. A. Gaby. Commenting on Mr. Darlington's statement, pointing out the advantages of intercommunication between electric systems to combine the loads into large blocks and permit the use at all times of the most economical available source of power: Ontario's smaller systems, operated by the Hydro-Electric Power Commission, are largely interconnected, and it is the purpose of the Commission to connect all systems (or at least the greater number), as soon as conditions warrant. For instance, the Eugenia, Severn and Wasdells systems are at present connected with the Niagara system, and it is the intention of the Commission to connect the Muskoka system with this group. This will afford considerable advantage to the smaller systems.

On the Severn system the hydroelectric generating plants have very little storage and are high load factor plants, whereas on the Eugenia system the hydroelectric developments are essentially storage plants, the power from these developments depending on the storage of the run-off of the drainage district in which they are situated, so by a combination of the two the maximum result can be obtained. That is, the off-peak periods of the Severn system are used to augment the storage in the Eugenia system.

Ultimately 12 out of the 14 systems will be interconnected for the purpose of taking advantage, not only of the variation in flow in the water power supplying the different generating systems, but also to take advantage of the difference in the peak-load period on the different systems. As the systems at the present time cover a territory of 800 miles east and west, the peak loads on the various systems, due to difference in time between the points situated at a distance from one another, do not occur simultaneously, resulting in improved load factor.

No. 1888

MODERN HYDRAULIC TURBINES OF LARGE CAPACITY

By H. G. Acres,[1] Niagara Falls, Ont., Canada
Non-Member

Modern super-turbine practice presents the problems of controlling high static pressures, of controlling and utilizing the flow characteristics in long pipe lines, and of controlling a highly energized water column and handling the heavy revolving weights of generator and turbine. The author outlines the general specifications for apparatus to accomplish these objects, and shows how the development of the Johnson automatic plunger valve, the Johnson differential surge tank, and the Kingsbury thrust bearing have successfully solved the problem and made the modern turbine possible. Other refinements in design which have had a notable influence on turbine development are improvements in turbine governors, experimental work on draft tubes, methods of preventing leakage, and the Taylor sectional scroll case. Pitting and corrosion of runners can now be avoided by the adoption of a conservative specific speed.

The improvement in efficiency of turbines is exemplified by the 55,000-hp. units of the Queenston plant of the Hydro-Electric Power Commission of Ontario. These have a maximum efficiency of 93.5 per cent, an efficiency at the point of maximum discharge of 88 per cent, and a capacity range of 37,000 to 60,000 hp. at efficiencies of 92 per cent or over.

New developments in test methods include the Gibson process of determining the rate of flow in long closed pressure conduits, and the Allen "salt velocity" method of determining flow, which is especially applicable to short conduits of non-uniform section. The Gibson method is more accurate and precise than the electrical measurements used in conjunction with it.

On New Year's Day of 1905 there was turned over in the plant of the Canadian Niagara Power Company, at Niagara Falls, the first 10,000-hp. turbine ever built. Almost exactly seventeen years later the first 55,000-hp. turbine ever built was

[1] Hydraulic Engineer, Hydro-Electric Power Commission of Ontario.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers. Slightly abridged.

turned over in the Queenston plant of the Hydro-Electric Power Commission. These two happenings mark an era of development which suggests an interesting subject for discussion, principally from the standpoint of the advances in the art that have made possible the construction and operation of turbines having capacities so greatly in excess of those which made history at Niagara less than twenty years ago.

2 For the purpose of placing a reasonable limit on the scope of discussion, a hydraulic turbine of large capacity may be defined as one of not less than 25,000 hp. capacity, and having a revolving weight, with the generator rotor, of not less than 200 tons, exclusive of hydraulic thrust. These limitations will eliminate any consideration of low-head installations and will give prominence to certain problems which become increasingly serious as the operating head ranges upward from 70 or 80 ft. As units of the class which has just been defined are rarely installed under heads as low as this specified minimum, the discussion is therefore more or less limited to medium- and high-head plants.

3 Where a high operating head is involved, there is always present the problem of controlling high static pressure in the interests of safety. In most cases there is also the problem of controlling and utilizing the flow characteristics of long pipe lines to insure not only safe but practicable operation. Finally, where a copious water supply is associated with a high head, there enter the problems of safely and efficiently controlling a heavily energized water column, and handling the immense revolving weight of the modern super-turbine.

4 Briefly, it may be stated that since the 10,000-hp. prototype of the modern super-turbine went into operation at Niagara eighteen years ago, the subsequent outstanding advances in the hydraulic art have been primarily in the direction of safer operation, not only from the personal standpoint of the operator, but from the impersonal and hardly less important standpoint of the investor and customer. Another point worth noting is that these advances, when they first materialized, had to do almost wholly with the contemporaneous conditions, and their possibilities as related to the present status of the art were more or less a matter of subsequent realization. In other words, these advances did not definitely anticipate the super-turbine, but their successful application to contemporaneous practice made its subsequent development technically and mechanically possible.

Static-Pressure Control

5 Control of static pressure involved in super-turbine water supply is a vital factor in relation to a maximum degree of safety and convenience in operation, and in continuity of plant output. Obviously the mechanism to exercise this function is

a valve, and such a valve, for super-turbine service, must have the following general specifications:

a It must be built in any size necessary to accommodate any conduit, however large.

b It must be designed to operate under any static head, however high.

c Its operation must at all times be safe, sure, and positive in opening under full static pressure and in closing under the most serious plant emergencies, such as the breakage of a wheel case or a runaway resulting from failure of governor control.

d It must open and close rapidly, and should have an automatic closing feature to meet the above-mentioned emergency conditions.

6 Such a specification was not met by any valve until the advent of the Johnson automatic plunger valve, which is now an indispensible adjunct to super-turbine installation and constitutes one of the advances in the general art which contributed to the possibility of present-day super-turbine development. As it now stands, the Johnson valve is an adequate mechanism embodying the ideal elements of simplicity, strength, and efficiency.

Flow Control

7 In the majority of cases the development of a high head involves the use of some type of closed pressure conduit of considerable length. In the case of an automatically governed plant where the ratio of head to length of conduit is more than about 1 : 10, certain phenomena become manifest which constitute a serious menace to safe and practicable operation. The continuous rejection and pulling on of load increments causes recurrent surges throughout the length of the conduit, which, if not damped out or relieved, will sometimes multiply and become superimposed to such an extent as to cause pressures enormously in excess of the static head. This condition, in conjunction with the conversion of pressure into velocity head for acceleration purposes when a large increment of load is pulled on, will impose a duty on the governors which they are not intended to perform, and where long closed conduits are used with turbines gated beyond the point of maximum efficiency, as they nearly always are, a condition may obtain which is entirely beyond the regulating function of the governor.

8 The power-discharge curve for the ordinary turbine will show a maximum production per second-foot at the point of best efficiency and the first derivative of this curve will show a rapidly decreasing rate of production per second-foot from that point to full gate. Consequently, every horsepower pulled on in excess of best-efficiency capacity will require a rapidly increasing amount of water to produce it. If, therefore, the

turbine is operating at or beyond the point of maximum efficiency when the system demands a large increment of load, there will be a falling off in production per unit of water supplied, and coincident therewith a loss of effective head, due to the absorption of such velocity head as is necessary to accelerate the water column. It is therefore evident that during the period when the water column is accelerating, a condition may obtain where the power output is actually falling off while the water input is increasing, the result being that the governor may open the gates full stroke in response to the falling speed, at which point its controlling function will cease, until either the system wr^2 holds the unit, with dropping frequency, for the space necessary for the governor to gradually resume control on rising speed, or until the generator drops its load and the turbine jumps to runaway speed at full gate, with possible disastrous consequences.

9 In so far as the control of surge pressure is concerned, it is evident that if no practical limitations are placed upon it in the matter of diameter and height, the ordinary standpipe would be an ideal corrective agency. It is clear, however, that in the case of a large-capacity high-head installation unrestricted scope in design is wholly impracticable from the standpoint of both cost and space limitation. Furthermore, a simple standpipe, feasibly designed and located, may at times actually make more acute the conditions it is designed to correct or alleviate, due to the fact it can only passively absorb surge pressure, with its recurrent phases. For this reason it often acts as an agency for superimposing, one upon the other, pressure waves generated by successive changes in load.

10 Another obvious expedient under this head is to limit the range of velocity change in the conduit by means of a synchronous bypass, actuated by the governor mechanism. Such a contrivance may be adjusted to prevent the occurrence of disruptive pressures in the conduit, and also to supplement the influence of the system wr^2 in holding the unit within the regulating range of the governor when the unit is pulling on or rejecting load, but only by reason of the fact that the condition under discussion has not necessarily to do with the maximum conduit velocity at any one time, but with the *range of change* in conduit velocity over a short period of time, and with the absorption and building up of head energy induced by these changes, within their low and high limits.

11 The synchronous bypass therefore has a useful function, but in the case of a high-head super-turbine installation its usefulness would be largely discounted by its cumbersome dimensions, its waste of water, and an added mechanical complexity which should be avoided whenever possible as a matter of principle.

12 The specifications of an ideal surge-control agency may now be defined as follows:

a It should have the effectiveness of a simple standpipe of very large dimensions, without its cost and space requirements.

b It should be capable on the one hand of preventing, or counteracting the effect of, and undue absorption of head energy when load is pulled on, and, on the other hand, of preventing, or counteracting the effect of, any serious recurrent surge pressures arising from load rejections, whether isolated or successive.

c The water surface should be sufficiently active to prevent freezing in ordinary low temperatures, and the dimensions should be such as to allow feasible and effective frost protection against extraordinarily low temperatures.

d It should operate without wasting water.

e It should be mechanically simple, with a minimum of moving parts and adjustments.

f It should be entirely dissociated from the governor mechanism, leaving the governor free to perform its own peculiar and highly important function, which is to control the speed of the generator, and not the vagaries of the water column.

13 This specification has been met with a large degree of effectiveness by the Johnson differential surge tank, one of the most useful and original inventions in the field of hydraulic engineering. Ample published data are available concerning its structural details and operating principle, and it is only necessary to state here that it constitutes another of the essential advances which have made possible the development and safe operation of the super-turbine.

Revolving Weights

14 The handling of the revolving weight of a modern super-turbine is governed largely by consideration of efficiency, a matter which will be discussed later, only the mechanical aspect of the problem being considered at this time.

15 In the early days of hydraulic-turbine development the step bearing, a form of combined thrust and guide bearing located under the runner, was used almost exclusively for the support of the revolving element. The subsequent gradual increase in the efficiency, speed, and capacity of turbine runners introduced problems of pressure intensity, depreciation, and accessibility, which were solved by abandoning the step-bearing principle and inaugurating the era of the horizontal-shaft turbine. This led to advanced development in bearings of the pillow-block type, together with the introduction of marine-type thrust bearings to take up unbalanced runner thrust, a double requirement which gave rise to serious problems when the development

of electrical generation and transmission called for continuously increasing turbine capacity and speed.

16 The subsequent development of the art under this head exemplifies a condition often met with in the field of engineering, where an inherently sound principle, after being applied in a primitive way, suffers a period of eclipse through arrested development, and perhaps years afterward is revived or rediscovered, and becomes the basis of the ultimate solution.

17 The original turbines installed in 1896 by the Niagara Falls Power Company represent the first reversion to the primitive basic principle. They were of the vertical-shaft type, the revolving weight being partially suspended from and partially superimposed on a step bearing located above the runner and immediately below the generator. This removed one of the main disadvantages of this type of bearing, that of inaccessibility. Another important innovation was the application of external pressure to a film of oil forced between the moving and stationary elements of the bearing.

18 Oil-pressure thrust bearings were in vogue for many years, and some are still in operation. Their disabilities are: high investment and maintenance cost, mechanical complexity, and high temperature, resulting in low oil viscosity and high energy losses. Also, even a momentary failure of the pressure-oil supply usually results in the loss of the bearing.

19 Finally there came the ultimate conception, based on the simple and true embodiment of a basic principle first established by the experiments of Tower in 1883 and mathematically demonstrated by Reynolds. Kingsbury exemplified this principle in a mechanism which is one of the outstanding examples of the simple and efficient application of a natural law.

20 The Kingsbury-type thrust bearing has the following characteristics which distinguish it from its oil-pressure prototype and make it an eminently suitable mechanism for supporting, in motion, the revolving weight of the modern super-turbine:

a It can be adapted to safely support the revolving weight of the heaviest turbine, within feasible dimensional limits.

b The source of oil supply is static, and integral with the bearing.

c The oil supply is "unlimited" in the sense necessary to conform with the laws of motion of viscous fluids as enunciated by Reynolds.

d The formation of the "pressure wedge" is not induced by any external agency but by the motion of the bearing itself, and by providing for a very slight lack of parallelism between the stationary and moving elements.

e The wedge pressure is a direct function of the speed of the moving element, and the thickness of the oil film is a direct

function of the degree of viscosity of the fluid, and consequently of fluid temperature, which can be regulated within any desired limits by the simple expedient of water-cooling coils.

21 Apart altogether from their very material contribution to operating efficiency, the mechanisms above discussed introduce a factor of safety and dependability, either singly or in combination, without which the present-day status of super-turbine development would have been unattainable.

Refinements in Design

22 In considering turbine governors under this head the question arises as to whether or not the governor should be classed as an essential advance in the art. The point is, however, that the items above classified as essential advances have been radical departures from contemporary practice and have actually opened up the possibilities of super-turbine development, whereas the turbine governor has been enlarged and improved more or less synchronously with the enlargement and improvement of the hydraulic turbine, and to meet the constantly increasing importance of the duty it has been called upon to perform.

23 *Governors.* As the primary requirements of governor operation are precision and reliability, the natural trend of design has been in the direction of simplicity in principle, the reduction of lost motion by the use of a minimum of moving parts, and precise shop work. An interesting development along this line is the White shaft governor, where the centrifugal element is attached direct to the main turbine shaft, thus obviating the complication of a belt or gear drive for the flyballs.

24 The double-compensation principle, with the restoring mechanism and load-limiting attachment, appears to meet all requirements of super-turbine governor control adequately and safely, and speed regulation is assured within safe operating limits for all gate movements up to the full stroke of the servomotors. Remote switchboard control has also been developed to such a stage that there is considerable latitude as to the location of the actuator, which in some cases is now placed directly over the servomotor cylinders, in others on the machine-room floor, and sometimes on the switchboard gallery.

25 A useful adaptation of the Johnson valve principle has been devised by Taylor for interchanging governor and hand control. Two sets of plunger valves are used, one set on the governor and one on the hand control, and the throwing over of one hand lever will simultaneously close all the governor valves and open all the hand-control valves, or vice versa. This operation can be carried out in such a short interval of time that there is little or no chance of losing control of the turbine,

and no chance whatsoever of interfering with the proper sequence of operation.

26 One of the latest innovations in governor design is a motor-driven centrifugal element, the motor being located on the flyball spindle and synchronously actuated by the main generator. This device serves the same purpose as the shaft governor in that it eliminates the belt or gear drive.

27 *Draft Tubes.* The activity which has manifested itself during the last four or five years in draft-tube investigation is directly traceable to the vogue of the super-turbine. There are two main reasons for this: First, the large concentration of power in a super-turbine permits an appreciable increment of useful power to be reclaimed as the result of a gain of only a

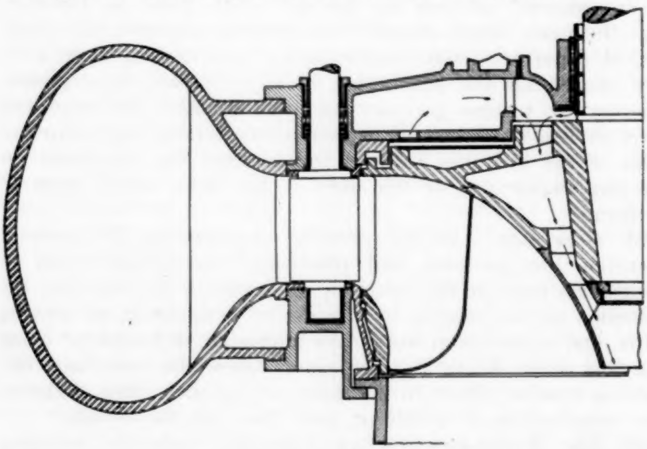


Fig. 1 Showing Application of Labyrinth Seal and Overn Disk to Hydraulic Turbine for Reduction of Leakage

fraction of one per cent in overall efficiency; second, the vibration resulting from vortex action and unsteady vacuum becomes magnified in the super-turbine to such an extent as often to cause serious inconvenience in operation. The pioneer development under this head was the White hydraucone, which marked a distinct advance in this branch of the art.

28 Briefly, the twofold object of recent draft-tube experimental research has been to regain as much as possible of the whirl component of the velocity head in the runner discharge, and to devise means of so training the combined axial and whirling flow, through appropriately designed water passages, as to minimize conditions giving rise to vibratory effects. All experiments carried on along this line by the principal turbine builders have definitely proved the inadequacy of all forms of the elbow-type tube, and have developed a series of types, more

or less correlated, which have given satisfactory results when tested under operating conditions. Of these types the Moody spreading tube seems to conform most logically to one's conception of the conditions to be met. This is particularly the case in respect of a cone in the center of the Moody tube which extends from the invert to the lower extremity of the runner hub. Inasmuch as this cone solidly fills the region of maximum turbulence, it seems entirely reasonable to assume, even without experimental confirmation, that it will be effective in preventing cavitation and the formation of vortices, with their resultant vibratory effects. Serving this purpose it also naturally follows that it must effect a useful conversion of the hitherto wasted

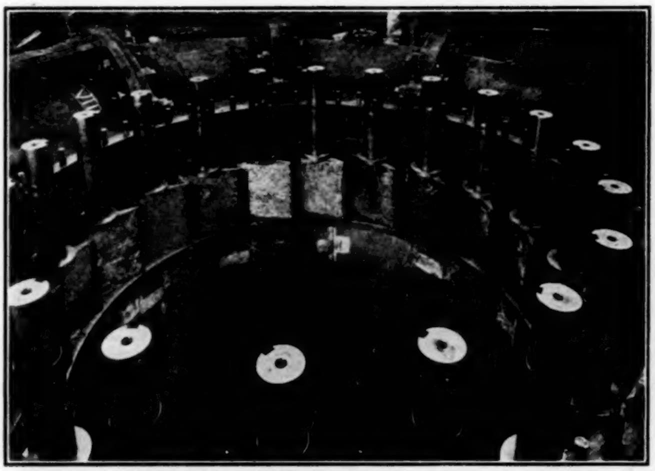


FIG. 2 SHOWING APPLICATION OF OVERN DISK TO GUIDE VANES FOR REDUCTION OF LEAKAGE

whirl energy of the central zone, realizing thereby a gain in efficiency as well as a betterment in operating conditions.

29 *Leakage Prevention.* In a small high-head plant with which the author had something to do, the turbine showed 85 per cent efficiency on acceptance test. A year later the efficiency had dropped to 67 per cent due to excessive leakage through the runner clearance spaces. The head in this case was 550 ft. and the circumference of the clearance spaces was about 14 ft. In the 55,000-hp. turbines at Queenston the head is about 250 ft. less, but the circumference of the runner clearances is about 65 ft. In the case of a runner of this size, under such a head, the leakage factor is a serious matter, and two of the most recent refinements in turbine design have been devised for the express purpose of meeting this condition: namely, the so-

called "labyrinth seal" for preventing leakage through the runner clearance space, and the Overn disk for preventing leakage through the gate clearances.

30 Instead of the ordinary simple seal consisting of a straight annular passage past the crown of the runner into the space under the head cover, and a similar passage past the runner band into the draft tube, the labyrinth seal, Fig. 1, consists of a series of alternately expanded and contracted passages which destroy the velocity head and reduce the head on the final free jet to one-third of its initial value.

31 The Overn disk, Fig. 2, functions by introducing into the clearance space between the end of the gate vanes and the

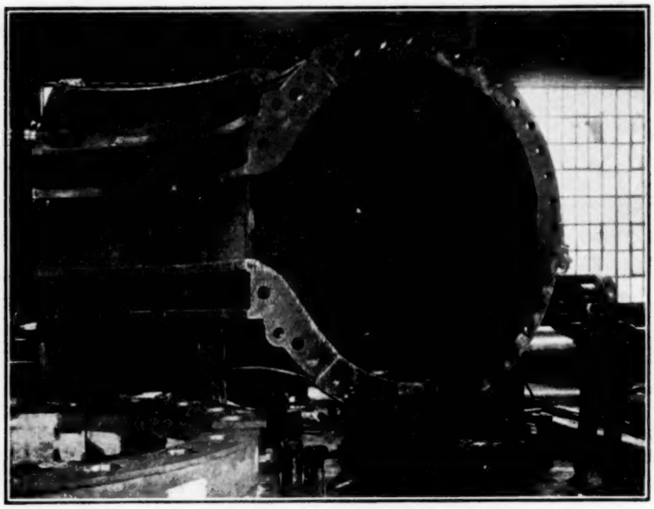


FIG. 3 SECTION OF TAYLOR SECTIONAL SCROLL CASE SHOWING INTEGRAL CAST STAY VANES AND JOINTS SUBDIVIDING SECTIONS

distributor plates an obstruction to the leakage flow considerably greater than the diameter of the gate shanks, thus effectively reducing the leakage at these points.

32 *Taylor Sectional Scroll Case.* Where the head is sufficiently high to require the use of a cast-steel scroll case, a serious problem is introduced which has to do with the cantilever strain on the radial joints between the speed ring and the scroll case. With the ordinary design efficient bolt distribution is not practicable, and in the shop pressure test, where the scroll is not supported by a surrounding mass of concrete, the bolts are sometimes stressed beyond the elastic limit, with resultant serious leakage. This disability can be largely overcome by casting

the speed-ring stay vanes integral with the scroll sections, but the pouring and annealing of such a casting is most difficult and the results are not always certain. The Taylor sectional scroll case, Fig. 3, is just as effective and largely removes any uncertainty as to the quality of the casting. The speed-ring stay vanes are cast integral with a small radial section only of the scroll, and connected later by means of a joint to the main section of the scroll. This joint is so located as to permit of a heavy flange and efficient bolt spacing, and the castings are of such shape and dimensions that there is every assurance as to their quality after leaving the annealing furnace. The showing made by the sectional scroll case under shop pressure test indicated that no undue risk would be involved in setting the scroll in the open, with only so much concrete as might be necessary to secure a stable anchorage.

33 With regard to the pitting and erosion of runners, super-turbine development has now reached a stage where this condition is no longer primarily a problem of design but of economics. If the customer so specifies, the manufacturer can select a specific speed and supply, at a price, a turbine in which the runner will have as long a period of useful life as the other major elements of the installation. Such a specification would, of course, involve additional capital expenditure for both generator and turbine, but as against this it must be realized that the modern super-turbine frequently has a capacity which enables it to earn upward of $2500 every twenty-four hours. Consequently the lost revenue charge against runner replacement in a fully loaded unit may easily run as high as $25,000. If this were necessary every two years, it would be equivalent, on a 6 per cent basis, to a capital charge of $200,000. Such being the case, it is evident that the choice of a proper economic specific speed is a matter deserving the most careful and mature consideration, being a factor of at least equal importance with proper gateage and elevation relative to tailwater.

Efficiency

34 The overall maximum efficiency of the Queenston units is well beyond 90 per cent, and within their maximum efficiency range they deliver about 32 e.hp. to the switchboard for every second-foot of water supplied under 305 ft. of net head. Under such conditions a variation of one per cent, one way or the other, on one of these 55,000-hp. units would mean either the lack, or the availability, of sufficient power to meet the requirements of an average community of 2000 population. This statement emphasizes the significance of high efficiency as related to super-turbine practice.

35 The primary factor making for high efficiency is a mini-

mum of obstruction to the direct flow of water from forebay to tailrace. This consideration involves water passages of ample section, with changes in direction of flow eliminated wherever possible, and where unavoidable, careful proportioning and transitioning. These requirements constitute the general definition of high efficiency and are exemplified in the carefully designed annular water passages in the Johnson valve, in the wheel case, speed ring, runner, and draft tube of the modern super-turbine, and also in the total elimination of one change in direction of

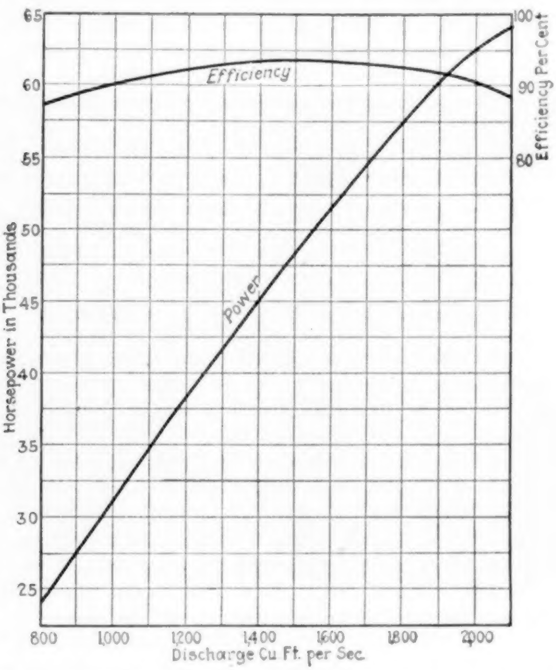


FIG. 4 POWER-DISCHARGE AND EFFICIENCY CURVES OF ONE OF THE QUEENSTON TURBINES

flow entering the wheel case, by vertically suspending the revolving weight from a Kingsbury-type bearing. Secondary but none the less significant factors are the elimination of leakage, low power loss in thrust and guide bearings, and expert shop work.

36 The proper handling and correlation of these various factors is exemplified in Fig. 4, which shows the efficiency and power-discharge curve for one of the Queenston turbines. The matters of interest in connection with these curves are:

1 The maximum efficiency is 93½ per cent.

2 The efficiency at the point of maximum discharge is 88 per cent.

3 The turbine has a capacity range of 32,000 to 63,000 hp. at efficiencies of 90 per cent or over, and a capacity range of 37,000 to 60,000 hp. at efficiencies of 92 per cent or over.

37 Fig. 5 shows the overall switchboard efficiency curve of the same unit, and indicates a maximum overall efficiency of 91 per cent. The outstanding fact in connection with these results

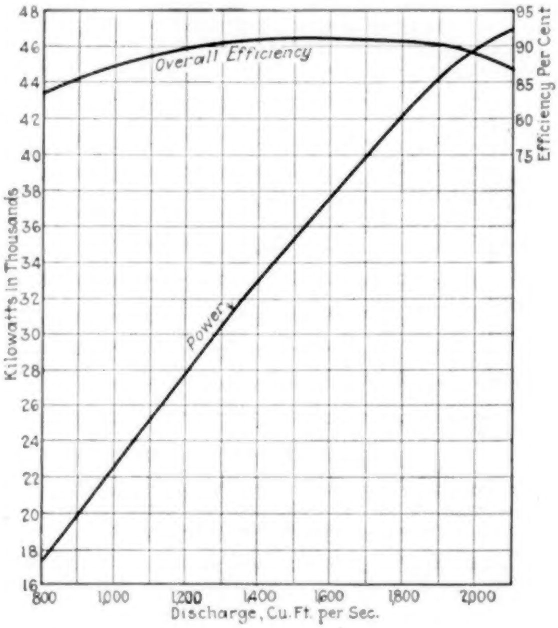


FIG. 5 OVERALL EFFICIENCY AND POWER-DISCHARGE CURVES OF UNIT OF FIG. 4; HEAD, 305 FT.

is that the modern super-turbine is capable of converting into mechanical energy all but 7 per cent of the gross potential energy of the water. Extreme conservatism in the fixing of specific speed, and the combined effect of further small refinements in the design of water passages, may possibly raise this efficiency another per cent in the future, and may give the curve a slightly more advantageous shape, but it would really appear that the super-turbine of the present day embodies the ultimate in respect of water economy at the point of best efficiency.

38 Fig. 6 represents the first derivative of the power-discharge curve of one of the Queenston turbines. As its name implies, this curve is derived by plotting the horsepower per second-foot

produced, in various regions of gate opening at and beyond the point of maximum efficiency, by a very small opening movement of the gates. It therefore shows the *rate of gain* in power for increasing water input. It will be seen that when operating at the point of maximum efficiency with a water input of 1600 sec-ft. the gain in power for one additional second-foot supplied is 32 hp. On the other hand, at full gate, when the turbine is taking water to the extreme limit of gate opening, the gain in power for one additional second-foot supplied is only about 10.5 hp.

39 The two extremes above cited illustrate the significance of statements made under the head of flow control, and also confirm the truth of the following: that under high heads,

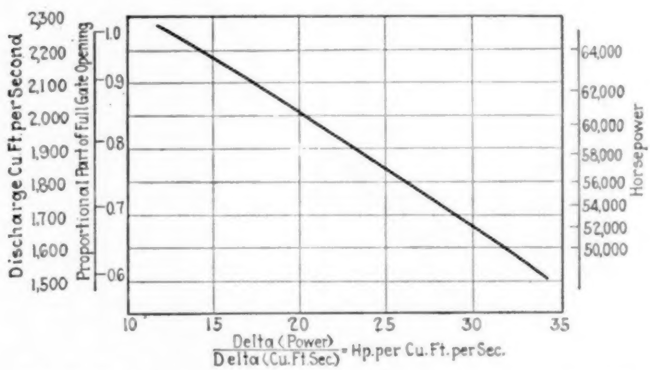


Fig. 6 First Derivative of Power-Discharge Curve of One of the Queenston Turbines, Showing Rate of Gain in Power for Increase in Discharge

where the gross potentiality per second-foot of water is correspondingly great, it is in the best interests of economy, as well as safety, to operate normally at the point of best efficiency, and to employ the excess capacity of an overgated turbine for emergency purposes only, and not for routine operation. This statement has greater significance when the water supply is artificially stored. Furthermore it is obvious that as the point of maximum efficiency is also the point of minimum hydraulic loss, the rate of runner deterioration at this point must also be a minimum. From this point on the rate of runner deterioration is an accelerating progression, directly related to the degree of overgate.

40 The matter of turbine gateage has not, in the past, received the attention it deserves from the purchaser. It does not particularly interest the manufacturer, and must be covered, if at all, by the customer's specifications. It would also appear that the main factor offsetting the above argument is one of cost and not of engineering: namely, the additional cost, if

any, of providing emergency plant capacity in the form of a separately installed unit or units, instead of relying on the overgate capacity of the normal installation.

41 It may be stated in conclusion that generalizations have probably less weight in hydraulic work than in any other branch of engineering. Nearly every prospective installation is a problem in itself, and no class of construction work is less governed by precedent.

Test Methods

42 The factors entering into a turbine test are forebay- and tailwater-level measurements, net head measurements, records of gate opening, and measurements of power and flow. Most of these factors are susceptible of easy and accurate determination by well-established methods, but in the case of flow the immense quantity of water taken by the modern super-turbine introduces an almost insurmountable problem as regards accurate measurement by ordinary means. However, there has recently been devised a method of flow measurement in closed pressure conduits which, by reason of its accuracy, cheap application, and conformity to an established principle of natural law, entitles it to be classed among the essential advances in the hydraulic art as related to super-turbine practice.

43 *The Gibson Method of Flow Measurement.* This method of flow measurement, known as the Gibson process, is based in principle upon Newton's second law of motion, and upon the less generally known theorem of Joukovsky, which is to the effect that when the velocity of flow in a closed pressure conduit is retarded an oscillatory pressure wave is induced, the intensity and amplitude of which is proportional to the degree of retardation and to the duration of the period over which the retarding influence acts. Briefly, Gibson used the penstock and turbine gates to produce the Joukovsky pressure wave, which he recorded graphically with an apparatus of his own devising and then reduced the result by the mathematical process of Newton's second law.

44 The method is therefore almost unique in the field of practical hydraulics in that it involves no empirical constants whatever, and the mathematical accuracy of the result is limited only by a relatively small instrumental and personal error involved in graphically recording and measuring the pressure wave. The Gibson process has been fully described in the technical press, and it is not the intention here to cover the details of its theory or application. It might be well, however, briefly to describe the method of obtaining and using the "pressure-time" diagram which forms the basis of computation.

45 The recording apparatus consists of a mercury U-tube connected to the penstock through a $\frac{1}{4}$-in. pipe. The glass leg

of this U-tube is connected to a camera box containing a lens focusing on a light-proof cylinder which carries a sensitized film and revolves at constant speed behind an oscillating seconds pendulum. When gate closure occurs, the mercury column in front of the lens rises and falls with the pressure wave, and the record is printed on the revolving film. At the same time the stem of the pendulum, swinging across the face of the lens, records the time period of gate closure on the film in the form of a vertical black line.

46 Fig. 7 illustrates a typical pressure-time diagram thus obtained. The point *A* represents balanced conditions in the system just previous to gate closure. The distance from *A* to the vertical line *EF* represents the full-time period of gate

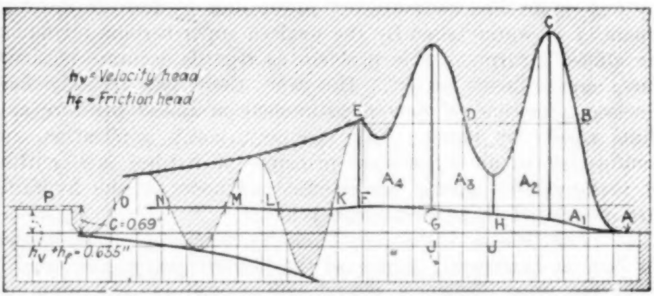


Fig. 7 Typical Pressure-Time Diagram Obtained in Flow Measurement by the Gibson Process

closure. The oscillations to the left of the line *EF* are due to the mercury column gradually stabilizing by friction, until at the point *P* static head is registered, with the penstock water column at rest. The vertical distance between the point *A* and the point *P* therefore represents, to the scale of the diagram, the recovery of velocity and friction head. The lines *J* represent the second intervals recorded by the pendulum.

47 The total area above the base line, and to the point of gate closure at *F*, represents a total energy absorption containing three separate elements, velocity head, friction head, and the destroyed momentum of the water column. This latter element being the one required for the application of Newton's second law, it follows that the energy absorption due to recovery of velocity and friction head must be segregated.

48 At the point *A* it is evident that the full value of velocity and friction head is existent, while it is equally obvious that at *F*, the point of final gate closure, this energy has been fully recovered. The line *AHGF*, technically known as the "recovery line," divides the total energy area into two parts, the upper

part of which, *ABCDEFGH*, represents the value sought, namely, the destroyed momentum of the water column. The area below the recovery line of course represents the sum of the other two elements, friction and velocity head. Intermediate points between A and P on this line are obtainable through the fact that the area generated at the end of any partial period of gate closure is proportional to the amount of flow reduction at that point, and also that the velocity and friction heads still unabsorbed are proportional to the square of the residual flow. The intermediate points G and H were determined on the basis of these relationships, through the medium of the measured sub-areas A_1, A_2, A_3, and A_4.

49 A test of one of the Queenston units, with all the other necessary operations in connection with head and gate-opening measurement, etc., was made complete within about four and one-half hours, and included 28 separate runs at various gate openings, an average of ten minutes per run. The unit itself was out of commercial operation for about one hour altogether, showing clearly the facility and cheapness with which the Gibson test can be carried out.

50 An interesting feature of these tests manifested itself in an apparent inconsistency in some of the finally computed points for the efficiency curve, which varied for equal power outputs. This trouble was due to a discrepancy in the power readings, which were actually correct in themselves within the error limits of laboratory standards. The reason is that under the Gibson process the velocity measured is that existing at the instant the gates begin to move, and if the speed of the unit is perfectly uniform at this instant, the corresponding power reading will be correct for the measured discharge. If, however, at the instant of the flow measurement the unit is in process of a speed change even so small as to be unnoticed on the frequency meter or tachometer, an error will be introduced in the electrical power measurement which the Gibson process is sufficiently precise to detect.

51 The flywheel effect of the Queenston generators is 21,000,000 lb-ft.^{2}. If, at the instant of a flow determination, the speed of the unit is dropping at even so small a rate as one cycle in ten seconds, there is 570 hp. registered on the wattmeters which the wr^2 of the generator is supplying, but not the water as measured.

52 Conversely, if the speed of the unit is in process of increase at the rate of one cycle in ten seconds, there is 570 hp. which is not being registered on the wattmeters, but which the water, as measured, is actually supplying, for absorption by the generator wr^2.

53 This means that for any one value of discharge there may be any number of wattmeter measurements which may

possibly be in error in any amount up to 1000 hp. It is possible that this disability may be overcome by devising some simple means of recording the minute speed change of the unit synchronously with the instant of flow measurement, and thus permit a proper correction to be applied to the power readings.

54 *The Salt Velocity Method of Measuring Flow.* The use of the Gibson process is naturally limited to conditions which are properly conducive to the production of the phenomena upon which the theory is based, the first essential being a closed pressure conduit of reasonable length. In short conduits, say, 50 to 60 ft. long, more especially if they have at the same time a non-uniform section, it becomes necessary to apply correction factors which have a greater relative influence on the ultimate accuracy of the result than is the case with long conduits. It so happens, however, that the recently devised Allen "salt velocity" method of measuring flow is susceptible of application to short as well as to long conduits and to non-uniform as well as uniform sections, the accuracy of the final result being dependent, not upon pressure rise, but on the refinement of method and money outlay applied to the injection of a salt solution, the time of passage of which, through a fixed length of conduit, forms the basis of the final computation. Practical applications appear to indicate that it has a useful function and will become a recognized process as related to this branch of hydraulic engineering.

55 In conclusion, it may be well to call attention to the fact that the definition of a turbine of large capacity, as given in the preamble of this paper, is applicable in particular to the Francis type, as there are now in operation, under heads of less than 70 ft., super-turbines which would come within the definition as regards capacity and revolving weight. These turbines are, however, of the new propeller type and any discussion of this new departure in turbine design would open up a subject far beyond the scope of this paper.

DISCUSSION

R. W. Angus. It appears from consideration of the present paper that much of what is said will apply with equal force to medium- as well as large-sized units, and much of it to low- as well as high-head plants. Surges are present in all plants having long, closed supply conduits, and the magnitude of the surge depends on the friction in the line and on its length and the initial and final velocities in the conduit, as well as on the diameters of the pipe and tank, but is independent of the head on the plant. The control of surges thus is really of more vital importance in low-head than in high-head plants.

Improvements in accessories have done very much to bring about the present advanced state of the hydraulic turbine, not the least important of these being the surge tank and the modern hydraulic valve. The perfection of the vertical thrust bearing has also been one of the greatest elements in the development of the vertical unit, which type of unit would otherwise have been impossible.

The author has spoken of the modern draft tube as an important device, and it has meant much to the large turbines, but as the paper excludes the low-head installation, it appears that the form of draft tube is not as important in high-head plants as we are often led to believe. In the modern low-head turbine of high specific speed the water enters the top of the draft tube with a high velocity of whirl, and the ordinary bent tube is unable to regain efficiently the energy in the water leaving the turbine. The water in such a tube whirls as it passes around the bend and the discharge conditions are bad. In this case the tube must have a special form, capable of recovering the energy leaving the wheel, and the hydraucone and spreading draft tube appear well suited to this purpose.

On the higher-head plants, wheels of comparatively low specific speed are always used, and it is relatively easy to deliver the water to the top of the draft tube in an axial direction at the load showing highest efficiency. If this is done the straight tapering tube, or even the quarter-turn tube, gives good results. It is true that the water will whirl at every load but one, and the spreading tube may show higher efficiencies on these loads than the quarter-turn tube does, but there is not yet enough published evidence in this connection to permit conclusions being drawn.

The author has given the results of tests on some of the wheels at the Chippawa plant, but does not state whether these are all for the one unit or not. The Chippawa plant was so designed that one turbine has the bent draft tube while others are made with the spreading tube, and if Mr. Acres would supplement his paper with test results on both types it would throw considerable light on the draft-tube problem.

Leakage is a serious matter in high-head plants and the author mentions two wheels, on one of which he has definite information about leakage loss. Judging by the head, the higher-head wheel would have a power of less than 5000 or 6000 hp. and the proportionate leakage in it would be much greater than in the Queenston wheels, so that, based on this illustration, leakage is relatively much more objectionable in the high-head, smaller units than in the larger ones. Water-turbine builders are trying the labyrinth packing, which has long been in use on steam turbines and turbine pumps. In both of these cases the rings have been used on smaller diameters than on water turbines and this fact makes the use of the labyrinth more problematical on water turbines than on

pumps and steam turbines, although it seems to offer a good solution to the leakage problem.

The author's tests and the efficiencies are most interesting and will be looked upon as indication of very marked advance in turbine design. The high efficiency of 93 per cent not only means efficient use of the water but longer life to the turbine, and the high efficiency over a wide range of loads is very desirable. Referring to the question of making reliable tests, there is serious danger of hurrying the work too much, and it does not seem desirable to shorten the tests unduly. Averages taken over a reasonably long time with steady loads would obviate the difficulty mentioned by the author, as he states that errors up to nearly two per cent are possible. I do not believe that the actual errors are as large as the author suggests are possible, but in any event they show that different methods of testing will produce different efficiencies on the same run, and the methods should be so standardized as to avoid this uncertainty of results.

H. Birchard Taylor. The Queenston project has been carried out substantially in accordance with the original program with respect to the unit capacity of the turbines. In the design of the turbines many new and interesting problems were involved due to their unprecedented capacity, to the large sizes of the castings required, and the unusual machine-shop processes involved in the construction of the turbines. Practically every improvement which has been developed in the art of turbine design and construction for high-head turbines has been incorporated in the units installed at Queenston.

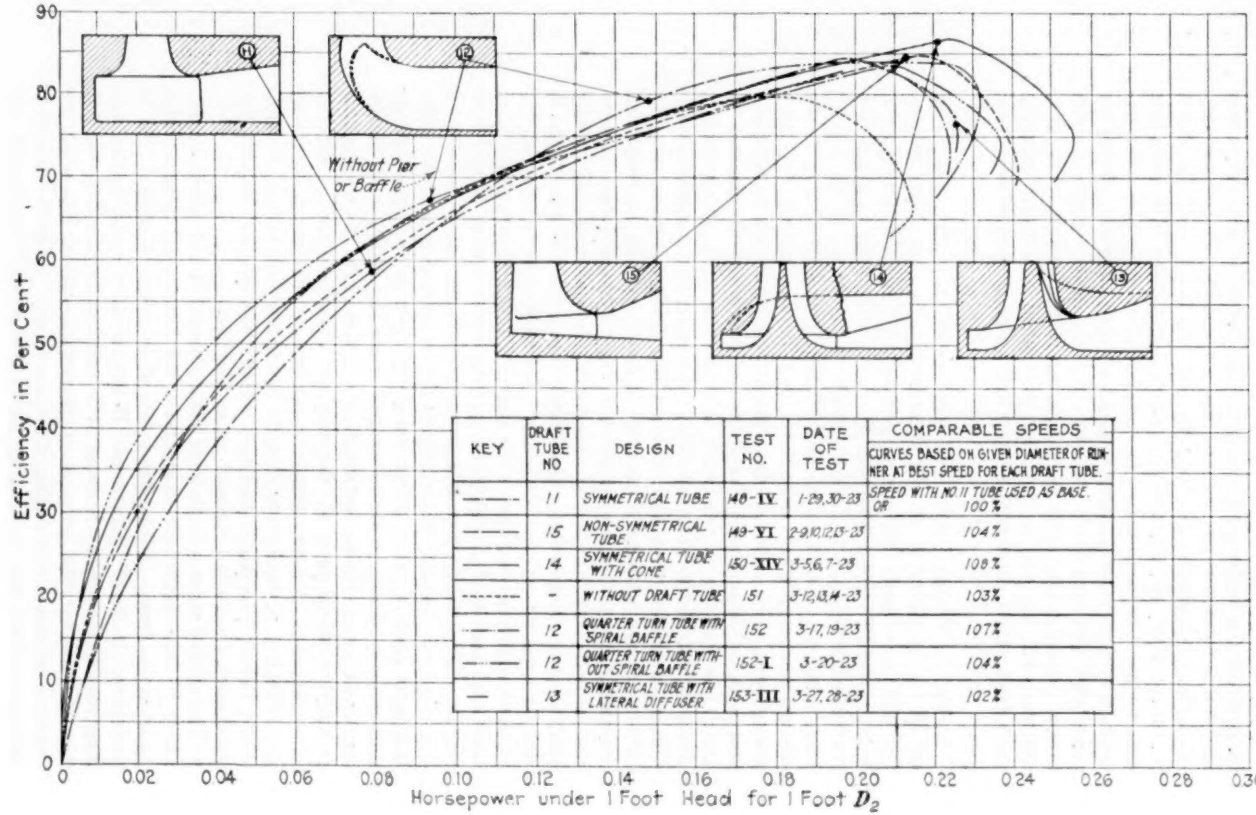
When units of such large powers are adopted, calling for the delivery of 55,000 to 60,000 hp. per unit, the importance of such questions as the attainment and continued maintenance of high efficiency, the elimination of corrosion of the runner, the avoidance of dangerous water hammer or vibration, the prevention of permanent injury from wear and tear by the provision of renewable parts, and careful study of operating requirements, becomes evident. The author's figures make it clear that an enforced interruption of service would not have to last many days to produce a loss exceeding the entire cost of a turbine unit. Any safeguards against interruptions of service either in the provision of auxiliary protection, such as the penstock valve, or in the construction of the turbine itself, will easily justify a large capital outlay.

When turbine units of 55,000 hp. capacity were suggested by the author in 1911, it was felt then that units of such capacity represented a radical step. Today we not only find these units in successful operation but there are now being constructed for the Niagara Falls Power Company units of 70,000 hp., and it would appear that the factors which will put a limit on the power

capacity of the turbine for developments being carried out in the future will be factors other than those involved in the construction of the turbine itself. Possibly the limiting factors will be found in the problem of continuity of service, but even in considering this factor it is reasonable to conclude that with the interconnection of power installations in great systems which is now taking place and will inevitably continue to take place, units of even larger capacity than any so far built, when feeding such large systems, will be just as dependable as units of smaller capacity have been when connected to systems of lesser magnitude.

Lewis F. Moody. Any one who investigates the manner in which the engineering of the Queenston plant has been carried out will be impressed by the purpose which has continuously been kept in mind by the author and his engineering staff to secure the best possible solution for every element of the plant design and by their adoption of new solutions for many problems arising in the design, whenever a new departure could be demonstrated to have advantages, even if its adoption was entirely contrary to accepted standards. There was well-established precedent for the use of many new features, for example, the Johnson valve, the spreading draft tube, and the Gibson method of testing the completed unit. Other features, such as Mr. Taylor's method of constructing the turbine casing without a separate speed ring, were worked out for the Queenston plant for the first time. Even in the features which had been previously tried out, however, an additional problem was introduced at Queenston due merely to the unprecedented capacity of the turbines. The Johnson valve required in these units was as large as the largest which had previously been built and had to be designed to operate under a head 50 per cent higher. The draft-tube designs had been previously developed but the responsibility placed upon the draft tubes in these units is greater than in others so far completed, just as the runner design and other hydraulic features assume increased importance due to the great amount of power involved. A difference of one per cent in turbine efficiency in all the units at Queenston means the loss of an amount of power greater than the entire output of one of the 5000-hp. turbines installed in the first plant of the Niagara Falls Power Company, turbines which far exceeded in capacity anything previously proposed up to that time.

The author in his specification for the penstock valve for installations of this magnitude enumerates the most vital points which must be covered. A supplementary feature which should also be considered is that of tightness. If the penstock valve passes any appreciable amount of leakage, it is unfitted for location close to the turbine for the purpose of permitting quick access to the turbine without unwatering the penstock. Another important feature is that the motion of the valve should not require the



KEY	DRAFT TUBE NO	DESIGN	TEST NO.	DATE OF TEST	COMPARABLE SPEEDS CURVES BASED ON GIVEN DIAMETER OF RUNNER AT BEST SPEED FOR EACH DRAFT TUBE.
—·—·—	11	SYMMETRICAL TUBE	148-IV	1-29,30-23	SPEED WITH NO 11 TUBE USED AS BASE. OR 100%
— — —	15	NON-SYMMETRICAL TUBE	149-VI	2-9,10,12,13-23	104%
———	14	SYMMETRICAL TUBE WITH CONE	150-XIV	3-5,6,7-23	108%
-------	-	WITHOUT DRAFT TUBE	151	3-12,13,14-23	103%
—··—··—	12	QUARTER TURN TUBE WITH SPIRAL BAFFLE	152	3-17,19-23	107%
—···—	12	QUARTER TURN TUBE WITHOUT SPIRAL BAFFLE	152-I	3-20-23	104%
—	13	SYMMETRICAL TUBE WITH LATERAL DIFFUSER	153-III	3-27,28-23	102%

FIG. 8

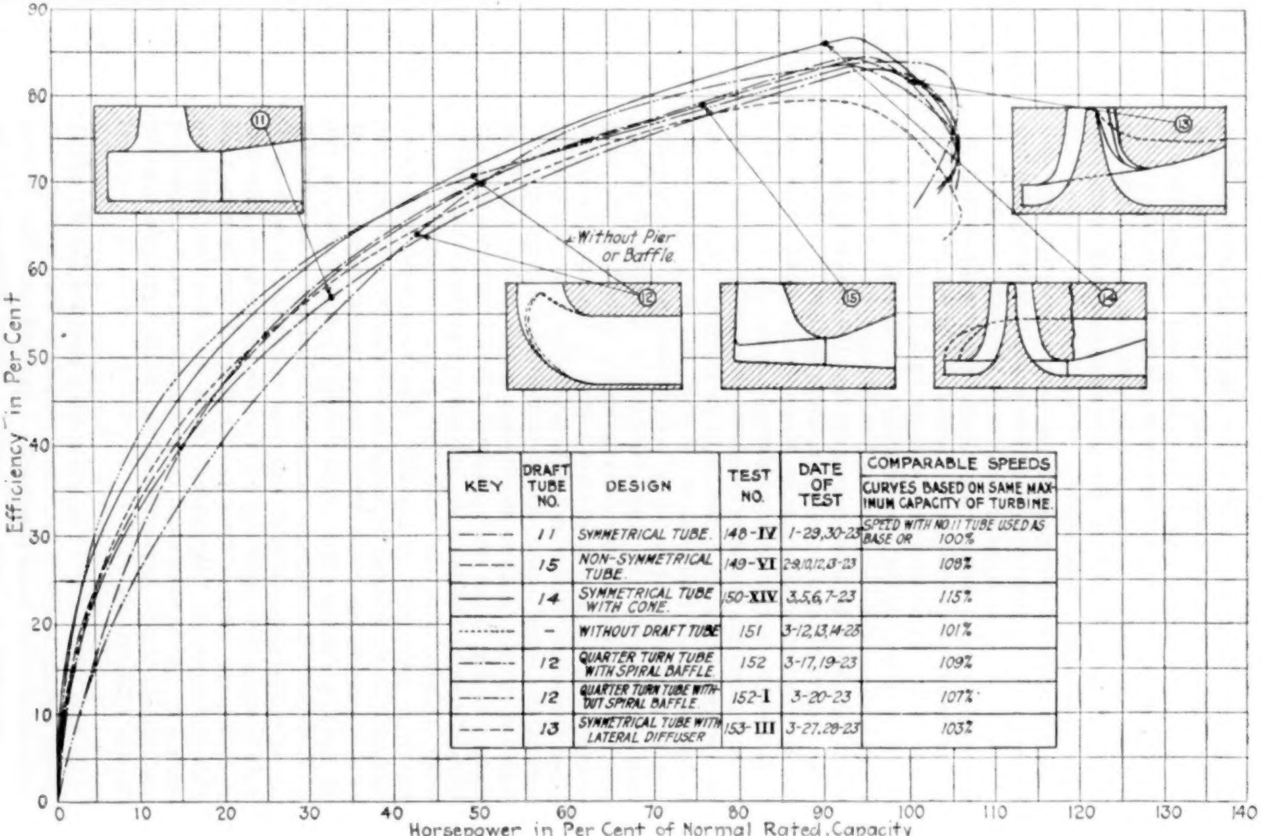


Fig. 9

Figs. 8 and 9 Turbine Tests Showing Results with Spreading Draft Tube Compared to Other Types

dragging of an unlubricated surface tangentially over another under a heavy normal pressure caused by the static head. The Johnson valve meets both of these additional considerations, as the seating takes place by a motion at a large angle to the seating surfaces, and only at the time of final closing does a heavy pressure due to the static head force the surfaces against each other, thus effecting extremely tight closure.

In connection with the author's reference to the design of spreading draft tube adopted for the last three units at Queenston in which a central core is extended all the way to the runner, it may be of interest to append hereto comparative tests recently carried out under the direction of Hugh L. Cooper & Co., for the U. S. War Department. These tests were made in the I. P. Morris laboratory in Philadelphia, under the direction of engineers representing Hugh L. Cooper & Co. and the War Department. The turbine was of very much higher specific speed than the units at Queenston and consequently involved a more severe draft-tube problem due to the relatively much greater tangential or whirl component present in the water leaving the runner. A draft tube which satisfactorily meets the requirements of this high-specific-speed turbine, however, will be equally applicable to the low-specific-speed turbine and its ability to regain energy of whirl will be effective in improving over-gate and part-gate performance in a lower-specific-speed installation. The particular form of spreading draft tube shown in these tests by the curve giving the highest efficiency was one in which the central core was carried to the runner as in the latest Queenston units. In Fig. 8 the efficiency has been plotted against the actual power output of the turbine when equipped with the various tubes shown in cross-section at the top of the figure. In Fig. 9 the same efficiencies have been plotted on the basis of the percentage of the full power capacity of the turbine when equipped with each tube, thus giving comparative results for equal percentage of full-load. The tests were undertaken primarily to see whether it would be possible by a simplification of design or the use of an elbow form of tube to reduce the cost of the power-house substructure sufficiently to balance the loss in efficiency. The increase in turbine efficiency secured by the use of the spreading type of tube is so substantial that its use is fully justified. It might also be of interest to mention that since the spreading form of tube was adopted for the Niagara Falls Power Company's "Hydraulic" plant and for the Queenston Station its use has continued until today it has been adopted for installations either built or building, which aggregate approximately 1,750,000 hp.

H. A. S. Howarth. The author has mentioned the Kingsbury thrust bearing as one of the radical departures from contemporary practice that has made possible the super-turbine. This bearing was developed by Professor Albert Kingsbury subsequent to his

study of Professor Osborne Reynolds' mathematical analysis of the experiments of Beauchamp Tower, which established that the lubricating oil film between two plane bearing surfaces must be wedge-shaped, and that in a journal bearing the oil film must be thicker where the oil enters than where it leaves. Professor Kingsbury had proved, experimentally in an air lubricated bearing, the existence of a continuous lubricating film that completely separated the journal from its bearing. He had measured the film thicknesses and pressures and found that the net pressure of the film balanced the load on the journal.

The Kingsbury bearing resulted from those studies and experiments. Its characteristic feature is the construction and mounting of one of the two bearing members so that the sections or segments of its surface automatically adjust themselves with relation to the other member so as to permit the formation of wedge-shaped oil films of the dimensions and proportions required by the operating conditions. The same mounting of the segmental surface that will serve best for one set of operating conditions of load, speed, and lubricant, will serve equally well for any other. The oil films begin to form under even very heavy loads, as soon as the bearing begins to turn, and complete films are established in a fraction of a revolution. The thickness of the film increases and its shape changes with increase of speed. By a suitable choice of lubricant and unit pressure the Kingsbury bearing can be successfully used for any speed and load, high or low, that is likely to occur.

The Michell bearing is of the Kingsbury type. It was independently invented by an Australian engineer, who also acknowledges his indebtedness to the mathematical treatise of Professor Reynolds. Although Professor Kingsbury's invention was several years prior to Mr. Michell's, the latter's British patent was issued a few years before the first Kingsbury patent in the United States.

William Monroe White. The writer is heartily in accord with the author's remarks on specific speeds. The insistence of electrical engineers on higher speeds means greater liability to pitting and corrosion. Mr. Acres is the first representative of the customer to say that the customer should also assume liability for some of the changes or damages resulting from higher speeds. The writer appreciates the author's reference to the hydraucone, and desires to pay a tribute to Mr. Moody's work. He desires to point out, however, that the hydraucone and the spreading draft tube are one and the same thing. Tests have shown that when the cone is taken out the results are the same. Mr. Moody's work, however, has shown that the cone steadies the performance of the unit, and the bringing up of the central cone to the hub of the runner may prove a valuable addition.

C. M. Allen. The salt velocity method of determining flow which has been mentioned by the author consists in introducing a

charge of salt water into a pipe line, and noting the elapsed time from its introduction until it leaves the pipe. It is then only necessary to divide the volume by the time to obtain the rate of flow, without any coefficients, factors, or corrections. By means of electrical recording devices, the passage of the salt solution can be accurately recorded, and if at the same time seconds are noted on the chart, all the events are recorded on the one chart.

The Gibson method, which uses the fundamental laws of momentum, is absolutely different but just as good. There have not been many opportunities to use both methods at the same time, but in the few cases where it has been possible they have absolutely checked each other. It should be impressed on every one who has to do with water power plants that if there is any apparatus on the job that can be used as a meter, it should be so used. The Johnson valve can be used as a meter by noting the difference in pressure head above and below the valve, and gives a first-class method of measuring water. It is just as important to measure the input into a plant as to measure the output.

THE AUTHOR. In connection with the remarks of Professor Angus as to low-head surges, it may be said that in low-head installations, surges and draft-tube conditions become major factors in design. As the scope of the paper was limited to high-head plants, it was impossible to give these subjects the discussion they would have merited in any discussion of low-head plants.

With regard to the question of error introduced into load measurements, it is quite true that the paper gives figures which indicate the possibility of a 2 per cent error. As a matter of fact, this phenomenon does not occur with every test run, and cannot occur when the load is really steady at the start of gate closure. It was merely mentioned to bring out the fact that in very large units there is no such thing as a "steady load," in the strictly academic sense; that is, where the load, over an appreciable period, corresponds exactly to any instantaneous value of the load during such period. The point is that the Gibson method was sufficiently precise to trace the error to its source, and to prove that it was due to minute speed oscillations and not to lack of refinement in velocity measurement, as might naturally have been assumed had one of the older methods been used.

In connection with Mr. White's comparison of the hydraucone and the spreading draft tube with the solid cone, what he says is substantially true with regard to the overall efficiency of a high-head unit, but from the operating standpoint, vibratory effects are now often a factor of primary concern. When Mr. White was carrying out his experiments on the hydraucone this was not the case, but with the advent of the super-turbine, and the attendant evolution of power-house design, the elimination of vibration and secondary resonant effects has become a serious problem of design and operation.

No. 1889

THE OIL VENTURI METER

By Ed S. Smith, Jr.,[1] Providence, R. I.
Junior Member of the Society

The paper presents a method of applying the venturi meter to the measurement of the flow of oil and other viscous fluids. It gives a chart showing value of the venturi-meter coefficient "C" over a range of turbulence from 0.0003 to 200,000, and flow graphs for a typical meter, by means of which the quantity flowing may be determined when a few easily measured constants are known.

THE venturi meter furnishes a means for accurately measuring the flow of liquids and gases in pipe lines within certain limits. These limits, however, must be known, and they are determined by the corresponding values of the turbulence, also known as Reynolds' criterion and defined by the method of dimensions as Qg/du. The theory underlying the method of calibrating the meter described below is not essentially new, but has been discussed by W. J. Walker and W. N. Bond as noted in references at the end of the paper.

2 In order to use the venturi meter with accuracy, the size and form of the tube must be known, and also the viscosity and density of the fluid. The several standard viscosimeters in common use in the oil industry determine the viscosity of liquids with commercial speed and sufficient precision.

3 The values of the coefficient of the venturi meter have been determined for a wide range of turbulences. The coefficient approaches unity at high turbulences but drops rapidly just above the upper critical turbulence and approaches zero as the turbulence approaches zero. This decrease of the coefficient is due chiefly to the increase of the friction pressure loss of the tube relative to the theoretical pressure drop, i.e., that pressure drop causing the increase of velocity from the entrance to the contracted throat of the meter.

4 Fig. 3 is an example of the most convenient form of cali-

[1] Assistant Mechanical Engineer, Builders Iron Foundry.

Presented at the Pacific Coast Regional Meeting of The American Society of Mechanical Engineers, Los Angeles, Cal., April 16 to 18, 1923.

bration for actual use, and is the result of two years' use of this method of calibrating the venturi meter with viscous oils.

Calibration by the Method of Dimensions

5 The increasing use of continuous processes in the oil-refining and other industries is bringing the venturi and other continuous flow meters into common use for the measurement of a large variety of fluids. As the viscosity of liquids may now be commercially determined, it is possible to use these meters with a method of calibration which is both simple and exact.

6 The method of dimensions has been in use for several years in the computation of friction pressure loss in pipe lines carrying viscous oils.[1]

7 The calibration has been extended to low values of the turbulence by employing data obtained by the author in tests conducted on a model venturi meter at the University of California. Grateful acknowledgment is made to Dr. Baldwin M. Woods, professor of aerodynamics, and to L. C. Uren, professor of petroleum technology, both of that institution, for their generous assistance.

8 The symbols and formulas used are as follows:

q = quantity, cu. ft. per sec.
Q = quantity, U.S. gal. per min. = $448.9q$
u/g = kinematic viscosity, sq. cm. per sec.
u = absolute viscosity, grams per cm-sec.
g = density (specific gravity, approximately), grams per cu. cm.
d = diameter of pipe, i.e., the same as the approach to the venturi tube, in.
a_1 = cross-sectional area of pipe, sq. ft.
a_2 = area of throat of venturi tube, sq. ft.
H = differential head of liquid in venturi tube from the approach to the throat, ft.
h = differential head of mercury in manometer measuring H, in.
C = coefficient for the venturi meter; see Equation [**1**]
P = friction pressure loss in pipe line per 1000 lineal ft., lb. per sq. in.
k = coefficient for friction pressure loss in pipe line; see Equation [**2**]

$$q = C \times a_2 \times \frac{a_1}{\sqrt{a_1^2 - a_2^2}} \times \sqrt{2 \times 32.2 \times H} \quad . \; . \; . \quad [\mathbf{1}]$$

$$P = k \times g \times \frac{Q^2}{d^5} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{2}]$$

9 Fig. 1 shows the value of the venturi-meter coefficient C as determined by actual experiments over a range of turbulence from 0.0003 to 200,000.

10 The friction-pressure-loss coefficient for pipe lines is also

[1] See The Friction Pressure Loss in Oil Pipe Lines, compiled by R. S. Danforth and published by the Kinney Mfg. Co., of San Francisco, Cal.

shown, the lower critical turbulence a occurring at 64 and the higher critical turbulence b at 85. From zero to 64 the flow is known as "viscous" or "streamline" flow, and the friction loss varies as the first power of the velocity. From 64 to 85 the flow may be termed "superturbulent," as the friction pressure loss varies as the cube of the velocity. From 85 to infinity the flow is known as "turbulent" or "hydraulic," and the friction pressure loss varies as a power of the velocity (the 1.75 power for smooth steel pipe lines such as are ordinarily used for oil transportation), with the square of the velocity as the upper limit for very rough pipe.

11 As the coefficient of the venturi meter is partly dependent upon the friction pressure loss along the venturi tube, it is ap-

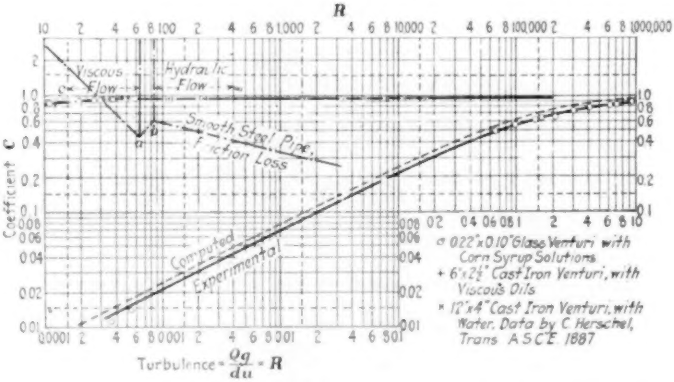


Fig. 1 Values of Venturi-Meter Coefficient C as Determined over a Range of Turbulence from 0.0003 to 200,000

parent that the law of variation of the coefficient with the turbulence will not be the same for the three types of flow and that consequently the graphical method of handling the coefficient is preferable to the use of an algebraic formula.

12 Fig. 2 provides an accurate calibration for the two types of venturi meters in common use in the United States, the Builders Iron Foundry and the Simplex Standard.

13 The Builders Iron Foundry tube has faired lines and apparently the higher coefficient. This calibration is for meters having a ratio of approach diameter to throat diameter of approximately 2.5 to 1, and is strictly correct for a 6-in. meter only — larger sizes having slightly higher coefficients and smaller sizes slightly lower coefficients. However, this calibration is accurate enough for oil measurement under commercial conditions.

14 The Simplex Standard tube has a ratio of diameters of 2 to 1, and all sizes are made practically similar in form. The coefficient is slightly lower than for the other type of meter,

but it is equally accurate as it is necessary with both types to use a calibration when measuring viscous oils.

Example in the Use of Fig. 2. Determine the quantity of oil flowing at 130 deg. fahr. through a 6-in. by $2\frac{1}{2}$-in. venturi tube (Type B.I.F.) when the mercury-oil differential $h = 4.2$ in., the kinematic viscosity $u/g = 0.11$ sq. cm. per sec. (from 64 sec. Saybolt), the density of the oil flowing is $g = 0.90$ gram per cu. cm. (from 26 deg. B.), and the density of the same oil at 60 deg. fahr. above the mercury in the manometer is 0.94 gram per cu. cm. from 20 deg. B. at this temperature).

Since $d_1 = 6$ in., $a_1 = 0.196$ sq. ft.; also, from $d_2 = 2\frac{1}{2}$ in., $a_2 = 0.0341$ sq. ft. Substituting these values in Equation [**1**],

$$q = C \times 0.0341 \times 1.016 \times 8.02 \times \sqrt{H}$$
$$= C \times 0.278 \times \sqrt{H} \text{ cu. ft. per sec.}$$

For a first approximation, assume $C = 0.95$; then

$$H = \sqrt{\frac{1}{12} \times \frac{13.6 - 0.94}{0.90}} \times \sqrt{h}$$

in which 13.6 is the specific gravity of mercury, and

$$Q = 448.9 \times 0.95 \times 0.278 \times \sqrt{\frac{13.6 - 0.94}{12 \times 0.90}} \times \sqrt{4.2}$$
$$= 0.95 \times 277$$
$$= 263 \text{ gal. per min.}$$

But $Qg/du = \frac{263}{6 \times 0.11} = 400$, and from Fig. 2, $C = 0.962$; therefore

$$Q = 0.962 \times 277 = 267 \text{ gal. per min.}$$

which is the quantity flowing under the conditions stated above.

15 Fig. 3 consists of Q-h curves for each of several viscosities of oil. The full lines are accurate to 1 per cent. The broken lines are in the viscous-flow region and are of undetermined reliability. The density correction for the head h is applied by the graph at the left side of the figure. A graph for the conversion of Baumé density to specific gravity (density in grams per cu. cm.) is attached to the upper side of the density-correction graph and shows the correction for the increase of density with a rise of temperature of the oil above 60 deg. fahr. At the right of the density-temperature graph is a small sheet of logarithmic cross-section paper for plotting viscosities at various temperatures of the oil which is being measured. The kinematic viscosity of ordinary petroleum oils, when plotted against temperature on logarithmic cross-section paper, forms a nearly straight line within the limits shown. As an example three plotted points are shown for a viscous California oil. The conversion from viscosity in seconds of flow with the Saybolt Universal viscosimeter to kinematic viscosity is shown on the upper margin of this graph. The viscosity of oil at any temperature, as shown on this graph, is used to indicate

the proper curve to use on the *Q-h* graph. This same group of auxiliary graphs may be used with any size of venturi meter, it being necessary to change only the *Q-h* graph.

Example in the Use of Fig. 3. Determine the quantity of oil flowing under the same conditions as in the example illustrating the use of Fig. 2.

Referring to Fig. 3, follow the arrowed dot-and-dash line for the 4.2-in. mercury-oil deflection from the left side of the density-head diagram to the 0.90 vertical density line, then up the diagonal to the corrected deflection $h = 4.68$ in. Follow this horizontal dot-and-dash line to its intersection with the diagonal 0.11 kinematic viscosity line (obtained by interpolation). Reading vertically down from this intersection it is seen that the quantity of oil flowing under these conditions is 267 gal. per min.

Limits of Accuracy of the Calibrations

16 The data submitted in Fig. 2 for the Builders Iron Foundry venturi tube are accurate to within 1 per cent for tubes of similar

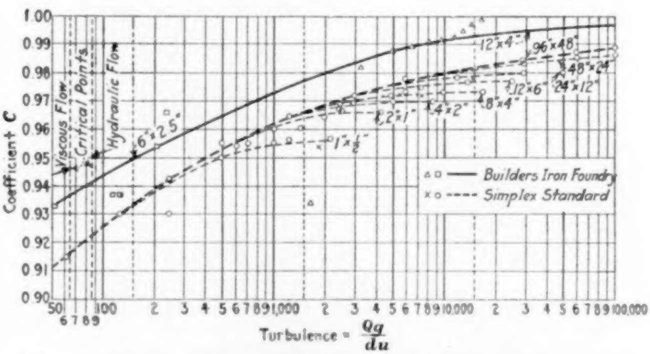


Fig. 2 Calibration for Two Types of Venturi Meters in Common Use in the United States

form (2.5:1 ratio of approach to throat diameter) for 2-in. to 12-in. diameter tubes in the region of turbulent flow, since the data in this region are calibrations of full-size venturi tubes with lathe-turned bronze or cast-iron throats.

17 This calibration applies strictly to 6-in. diameter tubes only; because the tubes to be similar in form must have the roughness increase directly with the size of the tube, i.e., the size of the rugosities must be in proportion to the diameter of the tube. The use of equally rough (or smooth) surfaces for all sizes of meters causes the coefficient to be slightly higher for larger meters for all turbulences in the region of turbulent flow. The degree of roughness does not appreciably affect the coefficient in the viscous-flow region.

18 The data in the viscous-flow region are not as accurate as those in the turbulent-flow region. The former are from a calibration made by the author on a 0.22-in. by 0.10-in. home-made

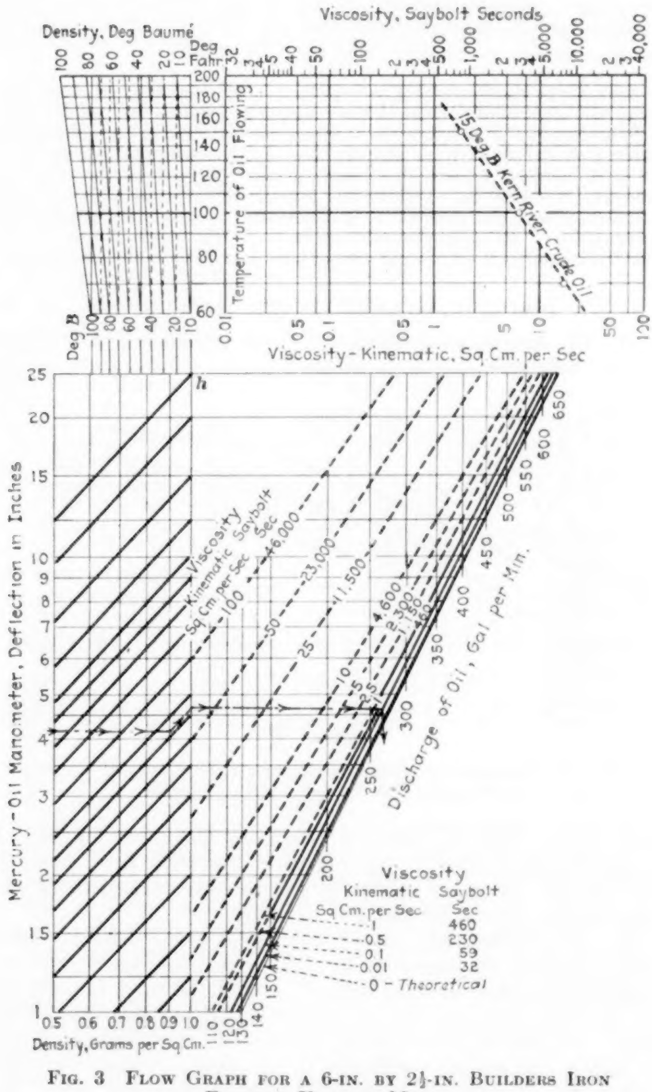


FIG. 3 FLOW GRAPH FOR A 6-IN. BY 2½-IN. BUILDERS IRON FOUNDRY VENTURI METER

model glass venturi-meter tube with water and corn-syrup solutions. The venturi tube was drawn from a straight glass tube and, while not exactly similar in form to the Builders Iron Foundry tubes, approached it closely enough to show definitely the type of variation of the coefficient to be expected for full-scale venturi tubes. The kinematic viscosity was determined with home-made pipette viscosimeters calibrated with water. In spite of the care used to avoid the effects of surface tension and velocity head, and because these viscosimeters measured solutions more than a thousand times as viscous as water, the probable accuracy of the viscosity determinations is approximately 90 per cent. This would mean an expected error of about 10 per cent in the turbulence for the values of the coefficient in the viscous-flow region.

19 For the purpose of checking the viscous-flow data obtained with the model glass venturi tube, the theoretical coefficient of a 6-in. by 3-in. Simplex venturi tube was computed. It was assumed that the only sources of loss of head from the approach to the throat were those due to the friction loss along the tube and the building up of the kinetic energy of the fluid in the throat. Poiseuille's formula for the friction loss of fluids in viscous flow in cylindrical tubes was used without modification for the approach and the throat. The friction loss in the cone was computed using the integral calculus and the same formula. From the sum of the change in the kinetic head and the friction loss the coefficient was computed and expressed by the formula —

$$C = \frac{1}{\sqrt{1 + (1.67/R)}}$$

These coefficients were plotted on Fig. 1. This calibration applies equally well to any other size of venturi tube of the same shape. It shows that the coefficient decreases indefinitely as the slope of the lower part of the line becomes the constant 0.5. The general agreement with the experimental data for the glass model, which was of slightly different shape, shows that the assumption made in this computation was justified for low values of the turbulence but is not sufficient as the critical turbulence is approached. This is indicated by the dropping of the experimental below the computed line, and is probably due to the increased internal losses in the fluid with the higher readjustments of velocity in the cone. At low values of the turbulence the venturi tube acts merely as a resistance and is equally as useful for the measurement of rate of flow as a piece of straight pipe with similar piezometer connections would be.

20 The data presented in Fig. 2 for the Simplex Standard venturi tubes are accurate to within 0.5 per cent for all values in the turbulent-flow region. This calibration is for all meters of 2:1 ratio of diameters for all sizes of tubes from 1 in. to 48 in. in diameter.

21 As liquids are usually transported at a much lower velocity (s) than the velocity (c) of sound in the liquid, the effect of the compressibility is negligible. The effect upon the venturi or pipe-line friction-loss coefficient due to the compressibility of the liquid flowing is found to be a function of s/c by the method of dimensions. When, as is sometimes the case, gases and vapors are handled at the acoustic velocity (c) or higher, the coefficients must be considered as functions of s/c as well as of Qg/du. Experimental data compiled on a graph according to this method would provide a rational calibration of venturi, orifice, and pitot meters for use with steam and high-pressure gas or air. The same method might similarly be applied to show the performance of steam-turbine nozzles and jet pumps.

22 The thin-plate orifice and fixed pitot-tube meters may be calibrated by the use of this same method of dimensions. The fixed pitot tube is very sensitive to irregularities of flow in the viscous-flow region, especially with heated oils; however, experimental data are necessary to determine its value and range of usefulness.

23 The accuracy of measurements by the venturi or thin-plate orifice meters in the viscous-flow region may be considerably affected by heated oil or by valves and fittings near the venturi tube or orifice plate, especially when the fittings are located on the upstream side. The amount of error due to these causes can only be determined by experiment.

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DISCUSSION

C. G. Richardson. The following idea is offered by F. N Connet. Consider a standard venturi tube, whose formula is

$$V = C\sqrt{\frac{r^2}{r^2-1}} \times \sqrt{2g(VH)}$$

where r = ratio of inlet area to throat area, and (VH) = the pressure drop from the inlet to the throat. In the measurement of water, the friction loss (FH) across the whole tube is a constant proportion of the venturi head (VH) and is approximately 0.1 (VH). In the flow of more viscous liquids, a certain proportion, greater than 0.1 (VH), would exist between the friction and ven-

turi heads. It is suggested that a series of tests be made with liquids of different viscosities over a wide range to determine, first, the value of C, and second, the relation between (FH) and (VH). If a constant ratio of (FH) to (VH) were found to exist for different viscosities, the addition of a third pressure at the outlet of the meter tube would provide a means of measuring all viscous fluids without determining their viscosities. That is, knowing this ratio, we would have the value C in the usual formula.

The Author. Replying to a question as to the effect of temperature on the coefficient C, temperature enters only to the extent that it affects the density and viscosity of the fluid, except in the region of viscous flow. In this region there is a slight effect upon the coefficient as the lines of flow are distorted. The chilled fluid near the sides of the pipe and meter tube flows more slowly on account of its increased viscosity, and also because the hotter liquid closer to the center of the pipe rises on account of its decreased density.

In reply to a question as to the relative accuracy of the venturi meter and a gaged tank, it may be stated that the indications of the meter are within one or two per cent of the actual rates of flow. In general, a continuous flow meter of the venturi or thin-plate orifice type is better than a gaged tank wherever it is necessary accurately to measure or control a liquid while it is flowing.

Commenting on Mr. Connet's idea, as presented by Mr. Richardson, the use of the ratio $(FH)/(VH)$ is an ingenious method of using the venturi tube as its own continuous viscosimeter. Fig. 2 clearly shows that the rate of flow affects the coefficient as well as the kinematic viscosity. Mr. Connet's idea, extended to include the velocity effect, if developed into a practical accessory to the venturi meter, would make it independent of external viscosity determinations. At present, however, it is necessary to measure the temperature of the fluid at the tube, and to use this temperature in connection with previously obtained temperature-viscosity and temperature-density data for the fluid measured.

No. 1890

THE CROSS-FLOW IMPULSE TURBINE

By Forrest Nagler,[1] Milwaukee, Wis.

Member of the Society

The author reviews the history of the development of the hydraulic turbine and shows that there is a gap between the field of the reaction turbine and the impulse wheel. This is due to the fact that the impulse wheel, as now built, has the slowest speed of all turbines, and that its highest specific speed is much below that of the reaction turbine. The cross-flow turbine, the principles of which are developed in this paper, will fill this field. The paper is not the presentation of a finished design, but rather the outline of the path that must be followed to meet new conditions in hydraulic power development.

The origin of the first prime mover is lost in antiquity, but we are able to state with practical certainty that it was an impulse-type water wheel; more specifically, that it was of the impact type, a current wheel with flat paddles, but an impulse wheel nevertheless. Written descriptions of these wheels date back nearly 2000 years, and current wheels, modified possibly to a slight extent in the direction of breast wheels, are probably several times that old.

2 The term "impulse" is commonly used with those wheels in which water is applied to the rotating element in a free jet with all its energy in the form of velocity. It will be so used here although the term is somewhat of a misnomer.

3 Modern impulse wheels develop power as much by reaction as they do by impulse or impact. The term "impulse" or perhaps better, "impact" might have been accurately used to designate the original current wheels and some of the pioneer forms, such as the hurdy-gurdy wheel, where the water was received on flat surfaces and no attempt was made to utilize the reaction of the water leaving the wheel. The modern wheel, however, receives the water without shock, completely avoiding what might be termed impact,

[1] Engineer, Hydraulic Department, Allis-Chalmers Mfg. Co.

Presented at a meeting of the Milwaukee Section, Feb. 20, 1923, and also at the Pacific Coast Regional Meeting of The American Society of Mechanical Engineers, Los Angeles, Cal., April 16 to 18, 1923. Abridged.

and turns it by a smooth path into a relative direction substantially contrary to that of the motion of the wheel. The essential distinction between the hydraulics of the so-called impulse wheel and the reaction type is found in the fact that in the former the entire energy of the water received by the wheel is in the form of velocity.

4 As this paper has to deal primarily with a modification of a single type of wheel or turbine the nomenclature of which is fairly well established, the term "impulse" will be adhered to as covering the general and specialized forms known variously as Pelton, tangential, Girard, Schwankrug, pressureless, and impulse or action types as contrasted to Francis, Jonval, Fourneyron, pressure, and reaction.

Historical

5 Prior to recorded history we may infer that mankind required a matter of centuries to produce the flat-blade impulse wheel. We know that this type persisted with little or no change, up to about 1500 A.D., that is, about two thousand years.

It required three hundred years more to get definitely beyond the flat-blade stage, realization of its disadvantages (only 50 per cent maximum theoretical efficiency) not materially affecting practice up to about 1850, although the faults of the flat blade were recognized 100 years previously. Some samples of the impact type of wheel survived in our own country up to the latter part of the nineteenth century, these being represented typically by the hurdy-gurdy wheel of the California mining days. These gave way to the impulse wheel in approximately its present form, Fig. 6 (*a*), during the period of 1870 to 1880.

6 As late as 1883 and again in 1890–1891 comprehensive tests made on impulse wheels included tests on designs of flat-vane wheels. Bulletin No. 1 of the University of California (June, 1883) is probably the impulse-wheel classic for all time. It sets forth clearly the reaction principle for all types, illustrates and analyzes arrangements of radial- and axial-flow circular-jet wheels, impact wheels with flat surfaces, single-lobe tangential wheels, and true splitter types. The mathematical analysis is exceptionally comprehensive without losing simplicity. Full appreciation of inherent disadvantages, such as "backed" water loss, was shown. Tests on impact wheels (40 per cent best efficiency) and Pelton wheels (82 per cent best efficiency) are given. Incidentally, this latter efficiency has not since been materially improved upon, considering the small jet size ($\frac{3}{8}$ in.) and low head (50 ft.).

7 Tests at the University of Michigan by Profs. M. E. Cooley, C. E. DePuy, and L. J. Hill in 1890–91 similarly included tests on both impact and impulse wheels of commercial forms and with modifications to improve the efficiencies of both. A maximum of 37 per cent with flat blades and 82.75 per cent efficiency with

Pelton buckets was obtained, the nozzle being $\frac{3}{8}$ in. in diameter and the head 92.4 ft. Analyzing the separate losses, the authors concluded that the bucket efficiency approximated 90 per cent, that the nozzle efficiency could be brought to 99 per cent, but that windage and friction losses could hardly be reduced below $3\frac{1}{2}$ per cent.

8 In the United States occasional patent references are available back to 1850, but in none of them is there a clear setting forth of the reaction principle as applied to impulse wheels nor a construction suitable for applying the principles until subsequent to that date. The earliest written account is that of Atkins, who applied for patents in 1853, although issue was delayed until 1875. Apparently Atkins fully appreciated the hydraulic principles involved.

9 It remained, however, for the mining industry of California to produce the predecessors of the type accepted universally at present. The wheels first used were known generally as the hurdy-gurdy wheels, and they dominated the period from 1850 to 1870.

10 The development work of the period 1870–1880 is dominated by the names of Knight, Moore, Hesse, and Pelton. The origin of the present type of tangential wheel, characterized by its cup-shaped bucket for securing full reaction and by its splitter for avoiding impact losses, lies with some or all of this group of men, but Pelton undoubtedly did most to develop and commercialize this form and to him belongs the credit for first increasing the efficiency to approximately where it stands today. Later modifications were made to improve the efficiency, to avoid erosion, or to secure some better mechanical standard, but the inherent characteristics have so far remained unchanged.

11 A most interesting historical account [1] and analysis of Pelton's work is contained in the report of a committee appointed by the Franklin Institute. This covered the various works, both European and American, leading up to Pelton's, disposed of contending claims, and finally made unqualified award on the basis of simplicity, economy of maintenance, adaptability to high heads, transportability, newness, correctness of principle, and commercial importance, but above all from the standpoint of efficiency. An appended test made at the U.S. Naval Academy in 1895 by Lieutenant F. J. Haeseler, U.S.N., and Ensign W. H. G. Bullard, U.S.N., shows a maximum efficiency of 86.56 per cent with a $\frac{3}{4}$-in. jet developing only 7.756 hp., volumetric measurement of water being used. This is the type that has remained unchanged to date.

12 European practice starting along a divergent line during the incubation period (1850–1880) of the American designs developed the Girard axial-flow and Schwankrug radial-flow impulse wheels for high head.

[1] *Journal of the Franklin Institute*, September, 1895.

13 Any working mechanism exposed to fluid in motion should desirably have the smallest possible hydraulic radius, that is, the least surface in contact with the water, to minimize losses and variations in velocity. For any given area the circle is the most advantageous shape, as no figure has a greater area for a given periphery.

14 The accident of circular jets used in our western mining work was responsible for a feature of design that was very instrumental in causing the American design of impulse wheel to supersede all others. This circular-jet nozzle, later of the needle type developed by Mr. Doble, obviated pitting trouble and low nozzle efficiency (frequently as low as 85 per cent) encountered in the various forms of partial annular nozzles, square nozzles, and tongue nozzles that characterized European design and were features of practically all Girard and Schwankrug wheels. Other than circular jets may work out advantageously, but so far all types of impulse wheels using them under high heads have failed or have been superseded commercially.

15 All impulse wheels have gradually resolved themselves into the specialized type known variously as the Pelton wheel, the tangential wheel, or, more generally, the impulse wheel. These have been highly improved, are very simple, efficient, and reliable, and gradually all other forms of pressureless wheels have given way to them. The last to go was the Girard impulse wheel or Girard turbine, as it is variously known. On account of its relatively large nozzle area it had a high characteristic speed and filled a gap not covered by either Pelton or Francis wheels except disadvantageously by the multiple-jet types of the former.

Characteristics of the Impulse Wheel

16 Practically all writers of hydraulic textbooks agree in their treatment of the impulse wheel that it is known by the following characteristics, the first three applying to impulse wheels in general, and the fourth identifying the present dominant type:

1 A free jet operating under the full spouting velocity due to the operating head
2 Substantially tangential application of the jet to the wheel, that is, with the major component of jet velocity along a tangent
3 A bucket velocity practically 50 per cent of the jet velocity
4 Splitter-type buckets, concave on the working surfaces.

17 According to definition, characteristic No. 1 is inherent with all impulse wheels. Nos. 2, 3, and 4 dominate the impulse-wheel field at present and have done so with almost no interference for thirty years, within which period practically all the development of modern water-power machinery has taken place. During the first half of this period these characteristics covered commercial

requirements which demanded the slowest-speed wheel that could be made under high heads. While certain conditions, notably high heads, still demand tangential wheels, for about fifteen or twenty years the requirements of commercial practice have frequently exceeded limitations imposed by characteristics Nos. 2, 3, and 4, and to meet these new conditions the new form of wheel proposed by the author has departed from these three characteristics in a decided and radical manner.

18 Contrary to the general impression, the modern impulse wheel is the slowest-speed type of turbine known, although it utilizes the highest water velocities. This low-speed type has found an extensive field of usefulness by reason of its mechanical simplicity and the relatively small surface exposed to high water velocities. Its low speed was essential in connection with the small capacities and high heads to which it was adapted, as it permitted revolutions per minute sufficiently low to readily permit of direct connection with alternators of reasonable speed. Its arrangement permits elimination of clearances and packing, which involve careful design and fine machine work and are a continual source of trouble under high heads. The small amount of surface exposed to water flow and the ease with which such surfaces may be inspected and renewed overcame one of the greatest problems in high-head turbine design.

19 It is probable that the intimate association of the low peripheral coefficient of 50 per cent with impulse-wheel design results from the fact that practically without exception authors of hydraulic treatises and writers of textbooks on the subject of hydraulic-turbine design confine themselves almost solely to this basis.

20 It is to this that the author takes exception, particularly since this 50 per cent basis is usually presented as a very fundamental consideration of all impulse-wheel design, whereas it really should be presented as an extreme used to permit of securing the lowest possible bucket speed. A single glance at the complicated forms of the most noteworthy impulse-wheel installation of the last few years should be all that is needed to indicate to the unbiased engineer whose ideas have not already been prematurely and positively fixed along a certain line, that the desirability of low speed in the buckets of these units has long since passed. The unit of Fig. 1 illustrates this point, as do also all of the record-capacity units above 5000 hp. or since 1904.

21 So far as water velocity is concerned, the impulse type of wheel should have the highest speed of any type of turbine, except those which may be classed within the so-called "suction" field. The water velocity in the usual reaction wheel (Francis) seldom exceeds 50 to 75 per cent of the spouting velocity, or the same percentage of that which holds in all impulse practice. The main reason for the low characteristic speed of impulse wheels is found

in the fact that the whole periphery of the Francis is utilized for developing powers, whereas the usual impulse wheel comes under the classification of partial turbine. The result is that in comparing on the basis of a certain horsepower the diameter of the impulse type becomes so large that its r.p.m. is unduly reduced. This is further emphasized by the fact that the reaction wheel usually runs with a coefficient of rim velocity in the neighborhood of 60 to 90 per cent as contrasted to the 50 per cent for the present impulse type.

22 Speeds of tangential impulse wheels cannot be increased by reducing their diameter beyond a certain limit illustrated in the lower curve of Fig. 2, because the bucket turns so abruptly

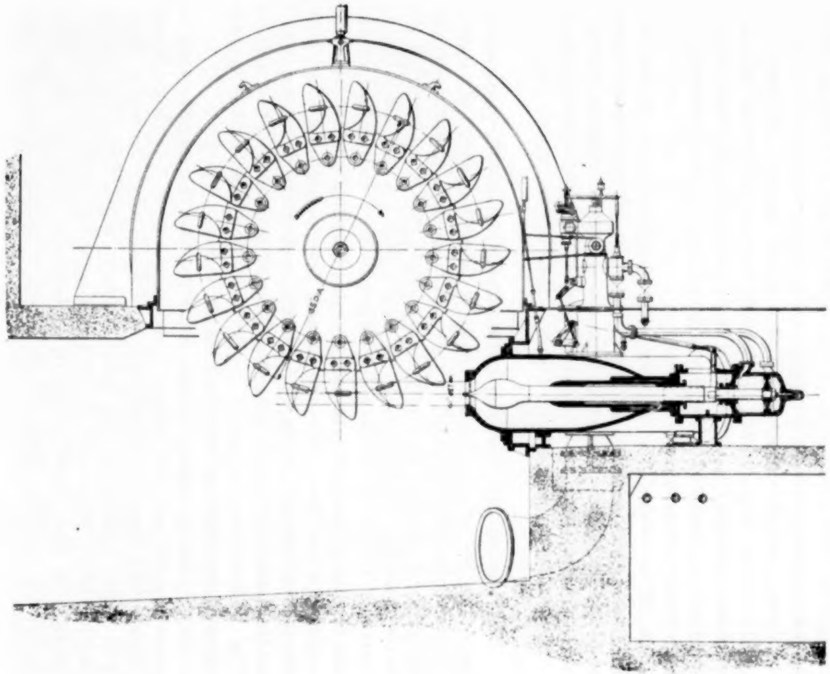


FIG 1. ONE OF THE TWO WHEELS OF THE LARGEST IMPULSE-WHEEL UNIT YET INSTALLED

(One of two units built in 1919 for the Caribou plant of the Great Western Power Co., California. Rated capacity of unit, 30,000 hp.; effective head, 1008 ft.; diameter of wheel, 155 in.; diameter of jet, 11 in.; speed, 171 r.p.m.)

out of the working path of the jet. The relationship between the characteristic speed of the tangential impulse wheel and its ratio of wheel to jet diameter illustrate the approximate limitations of commercial practice. Exceeding the ratios indicated simply means that there will be what is called "racing water," with its inherent loss of efficiency.

23 Unit capacities have so grown that for a considerable period, possibly for the last 15 or 20 years, strenuous efforts have been made to increase specific speeds of impulse units. As is usual in such cases, human inertia has been such that the new need was not recognized, the result being that the art of impulse-wheel design went through the same evolution as did that of

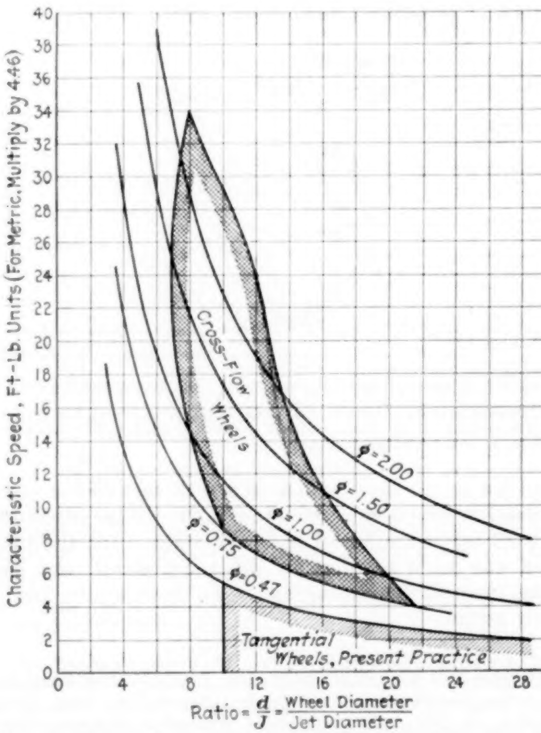


FIG. 2 CURVES SHOWING RELATION BETWEEN CHARACTERISTIC SPEEDS OF IMPULSE WHEELS AND THEIR PROPORTIONS AS EVIDENCED BY THE RATIO d/J

(The plottings for the various peripheral coefficients indicate the increases possible with larger coefficients than are used at present. The lower shaded area indicates the approximate limits of proportions and characteristic speeds of tangential impulse wheels, and the upper area roughly the field available for development by the cross-flow wheel. The increase in characteristic speeds that is possible is quite strikingly shown.)

reaction-type turbines, i.e., multiplicity of runners, nozzles, etc., without attacking the fundamental element of relative bucket velocity or direction of jets on the wheel to avoid interference. The orthodox 50 per cent velocity of bucket and tangential application of jet with its large consumption of wheel periphery were and still are the dominant factors of the impulse units of even recent manufacture and of all engineering textbooks on the subject.

Experimental Work of the Author

24 During the years 1913 to 1918 the author was experimenting almost continually with flat-angle runners of the axial-flow type. This work resulted in the development of the high-speed suction type of propeller runner since applied extensively to low

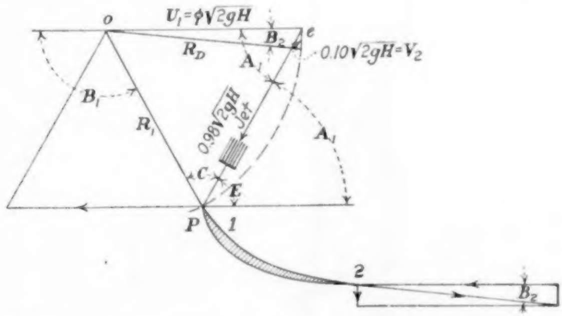


FIG. 3 FUNDAMENTAL DIAGRAM OF THE IMPULSE WHEEL

(Diagram is laid out for the axial cross-flow type of wheel, but represents an average of all types, whether of radial-inward, radial-outward, conical, or axial arrangement.)

heads and led to an appreciation of the characteristics necessary to high speed in any vane moving in a fluid. In his paper describing this work before the A.S.M.E. in December, 1919, the following statement was made as covering any water wheel, the ice-boat illustration incidentally coming under the classification "impulse" as defined herein:

> Neglecting friction and possibly blade thickness, there are no mathematical or hydraulic laws that will prevent doubling or quadrupling any particular characteristic speed by simply flattening the blade angles. A direct analogy to this is the well-known illustration of relative velocities evidenced in the sail of an ice-boat.

25 Continuing this work on high-speed runners with impulse wheels, the author in company with J. F. Roberts experimented with small-angle single-lobe types of buckets with the jet making small angles with the tangent. The purpose was to secure an arrangement whereby more jets might be used on a single wheel

without interference and to eliminate the splash losses resulting from water discharged upward and falling back on the wheel in vertical shaft arrangement. The result was difficulty from the "backed" water indicated at B in Fig. 6, which was one of the causes of the low efficiency and pitting that contributed largely to the commercial failure of Girard wheels and other single-flow impulse turbines. In playing jets from an ordinary garden hose on small models of these wheels that were running at a high rate of speed (10,000 to 20,000 r.p.m.) a decided change in their tune was noticed with various positions of the jet. On account of ease in construction one of the wheels happened to have been cast

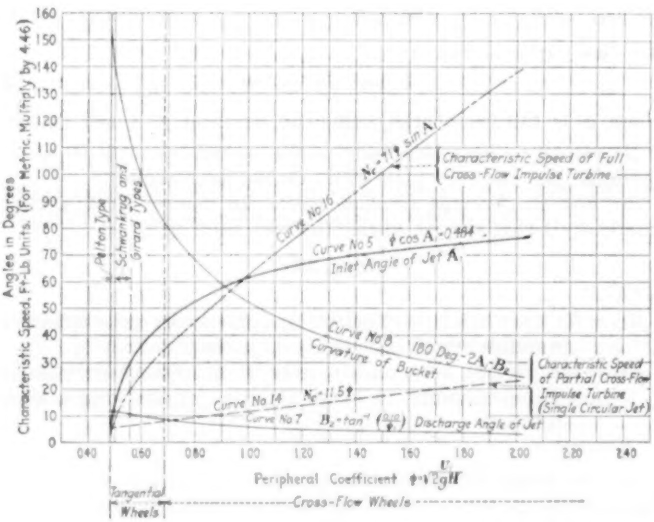


FIG. 4 DIAGRAM OF THE ENTIRE FIELD OF IMPULSE WHEELS, SHOWING THE GENERAL CHARACTERISTICS OF MODERN TANGENTIAL IMPULSE WHEELS OF THE SUPERSEDED HISTORICAL TYPES AND OF THE PROPOSED NEW CROSS-FLOW TYPE

(Assumptions: Discharge loss = 0.10 $\sqrt{2gH}$; jet velocity = 0.98 $\sqrt{2gH}$; overall efficiency = 80 per cent; d/J = 10.)

without back curvature of buckets, similar to the propeller-type suction runner. With this wheel the highest pitch or note was developed with the jet directed almost perpendicular to the plane of the wheel. This was so contrary to fundamental impulse practice as indicated by all commercial installations built in recent years and covered in any treatise on hydraulics that it attracted instant attention. Analysis of the condition resulted in a series of diagrams such as Fig. 3, and these worked naturally into the following algebraic analysis.

Referring to Fig. 3, let

$$\text{Jet velocity } V_1 = 0.98\sqrt{2gH}$$

From the right triangle on base *oe*

$$R_D = \sqrt{U_1^2 + V_2^2}$$

and from the law of cosines

$$R_I = \sqrt{V_1^2 + U_1^2 - 2V_1U_1 \cos A_1}$$

But $R_D = R_I$,

$$\therefore U_1^2 + V_2^2 = V_1^2 + U_1^2 - 2V_1U_1 \cos A_1$$

whence

$$\phi^2(2gH)+0.01(2gH)=0.96(2gH)+\phi^2(2gH)-2\,\phi\times 0.98(2gH)\cos A_1$$

and (see curve 5, Fig. 4)

$$\phi \cos A_1 = 0.484 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where ϕ is the peripheral coefficient, or ratio of wheel velocity to jet velocity. Note that varying the assumption of $0.10\sqrt{2gH}$ discharge loss has very little effect on the inlet angle.

26 For the average impulse wheel the bucket inlet angle is practically twice the jet angle, that is (see Fig. 3),

$$B_1 = 2A_1$$

Proof:

$R_D = R_I$ since relative velocities are practically constant in an impulse wheel (exactly so in axial flow).

$R_D = oe$ (Fig. 3) very closely since B_2 is always very small. Exact equality could be attained by allowing a slight forward component.

$oe = oP$

$\therefore$ Angle C = Angle A_1

But $A_1 = E$ by construction and $C + E = 2A_1$

$\therefore B_1 = 2A_1$

From right triangle,

$$B_2 = \tan^{-1}\left(\frac{0.10\sqrt{2gH}}{\phi\sqrt{2gH}}\right)$$

$$= \tan^{-1}\left(\frac{0.10}{\phi}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

$$\text{Bucket curvature} = 180° - B_1 - B_2 = 180° - 2A_1 - B_2$$

$$= 180° - 2\cos^{-1}\left(\frac{0.484}{\phi}\right) - \tan^{-1}\left(\frac{0.10}{\phi}\right) \quad . \quad . \quad [3]$$

(See curve 8, Fig. 4.)

27 With impulse wheels simple relationships may be deduced for comparative purposes as follows, letting J = jet diameter or thickness and d = wheel diameter, both in feet.

28 The equation for the characteristic speed of the wheel is —

$$N_e = \frac{\text{r.p.m.} \times \sqrt{\text{hp.}}}{H^{\frac{5}{4}}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [4]$$

and the speed is

$$\text{r.p.m.} = \frac{60\,\phi\,\sqrt{2gH}}{\pi d} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

Assuming 80 per cent efficiency and one circular jet per wheel

$$\text{hp.} = \frac{\pi J^2}{4} \times \frac{0.98\sqrt{2gH} \times H \times 62.4 \times 0.80}{550} = \frac{J^2 H^{\frac{3}{2}}}{1.78}$$

and

$$\sqrt{\text{hp.}} = \frac{J H^{\frac{3}{4}}}{1.334} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

Substituting [5] and [6] in [4] gives

$$N_e = \frac{115\,J\phi}{d} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7]$$

Most engineers dealing with the subject are familiar with Equation [7] expressed for tangential wheels as

$$N_e = 55\,\frac{J}{d} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

ϕ being taken as equal to 0.48.

29 Modern tangential impulse wheels preferably have a ratio of wheel diameter to jet diameter of 12 to 14, with 10 as a desirable minimum. Using the latter value for comparative purposes,

$$N_e = 11.5\,\phi \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

for impulse wheels with a single circular jet (see curve 14, Fig. 4).

With multiple jets the limit of characteristic speed is reached with a solid annular jet of width J or when the nozzle area is πdJ. In this case, assuming 80 per cent efficiency,

$$\sqrt{\text{hp.}} = \frac{2H^{\frac{3}{4}}\sqrt{dJ}}{1.334}$$

which, substituted with [5] in [4], gives

$$N_e = 225\,\phi\sqrt{J/d}$$

neglecting for the moment the jet angle; and for the d/J ratio of 10 assumed above,

$$N_e = 71\,\phi \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [10]$$

30 Since the full annular area of the nozzle is not effective on account of the inlet angle of the jet expressed in [1], Equation [10] must be altered to read

$$N_e = 71\,\phi\sqrt{\sin A_1}$$

(see curve 16, Fig. 4), since the quantity and consequently the hp. vary as the axial or radial component and N_e varies as $\sqrt{\text{hp}}$.

31 The significant relationship between wheel speed and jet angle is shown perfectly in Equation [**1**], the graphical showing being curve No. 5 of Fig. 4.

32 Considering the fact that this relationship is based on the fundamental requirements of an impulse wheel, i.e., low exit loss and constant relative velocity (resulting from no change in pres-

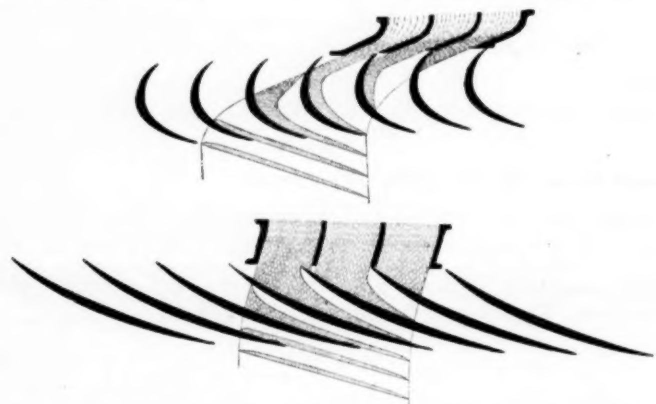


FIG. 5 DIAGRAMS ILLUSTRATING THE RELATIVELY SMALLER PORTION OF THE PERIPHERY OF A WHEEL UTILIZED BY A GIVEN JET OF THE CROSS-FLOW TYPE AS CONTRASTED WITH THAT OF THE MORE NEARLY TANGENTIAL FLOW. THE POSSIBILITY OF UTILIZING A LARGER NUMBER OF JETS WITHOUT INTERFERENCE IS EVIDENCED

sure) in the bucket itself, the curve may be used for some very broad conclusions.

33 The main one is this: *Speed in impulse-wheel work is a function of the angle the jet makes with a tangent.*

34 If the jet angle is zero as in accepted present practice, the speed, which is a function of ϕ, is the lowest that can be obtained without preventable loss. As the angle of the jet increases to 45 deg. the relative wheel velocity increases from 50 per cent to only 69 per cent of the jet velocity. From this point, however, the most radical increase is effected. For example, from a 69 per cent coefficient at 45 deg. the speed increases to 100 per cent at 61 deg. This feature combined with the obvious departure from the tangential flow of present practice led to the following designation of two classes of impulse wheels accordingly as the major component of the jet velocity is along or normal to the tangent:

Tangential Impulse Wheels — Jets making angle less than 45 deg. with tangent.

Buckets decidedly concave, curvature greater than 90 deg.

Bucket inlet inclined backward.

Cross-Flow Impulse Wheels — Jets making angle greater than 45 deg. with tangent.
Buckets flattened, curvature less than 90 deg.
Bucket inlet inclined forward.

35 The name "cross-flow" was selected after considering such terms as "vortex impulse" or "annular impulse" (on account of the ease with which so many jets may be used as to form a continuous whirling jet); and "axial impulse," "radial impulse," "mixed-flow impulse," etc., for reasons connected with the noticeable feature of various mechanical arrangements. "Cross-flow" seemed especially appropriate not only because it so aptly describes the direction of flow across the wheel but because it so distinctly defines the direction of flow as contrasted to that in the present wheels which are universally known as the tangential type.

Advantages of the Cross-Flow Type of Impulse Wheel

36 The cross-flow wheel, aside from its advantage in speed, automatically corrects one of the defects that contributed greatly to the failure of single-flow impulse turbines such as the Girard and other partial wheels. The "backing" of water in the wheel, resulting in its being dragged around with the wheel and ultimately discharged at wheel velocity at about half its original velocity, results from impact due to the large angle between the relative water path and the surface of the bucket at the point of impact [point *B*, Fig. 6 (*b*)]. The slower the speed of the bucket, the greater the curvature and consequently the greater the impact and "backing" loss.

37 With higher speeds the bucket becomes flatter and the "backing" loss with its poorer efficiency and greater tendency to pit is more and more reduced [see *B*, Fig. 6 (*d*)] without resorting to the undesirable expedient of increasing the number of buckets. Inspection of the successive diagrams of Fig. 6 indicates how the angle of impact between the jet and bucket is successively reduced as the speed is increased and the bucket correspondingly flattened. The author believes that this principle can be utilized to eliminate or at least reduce the efficiency losses and pitting troubles experienced with the Girard types, and with it there will be a rapid return to radial and axial and even conical or mixed-flow impulse wheels. In combination with nozzles designed to deliver circular jets with their higher efficiencies and more uniform distribution of velocities, cross-flow wheels can be applied to a considerable portion of the field for which there is at present no design except the disadvantageous multiple-runner or multiple-nozzle types. For low heads and small powers or where efficiency may not be of the greatest importance, the circular jet may be dispensed with and rectangular jets arranged to partly cover the wheel periphery (partial turbines), or even solid annular jets (full turbines) may be utilized.

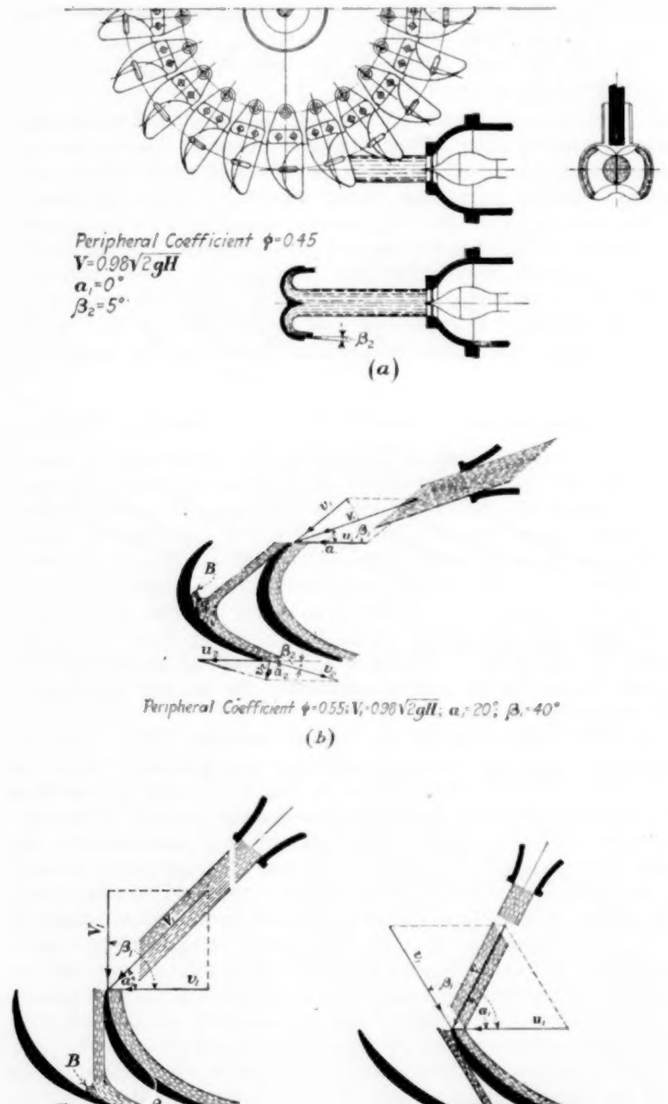
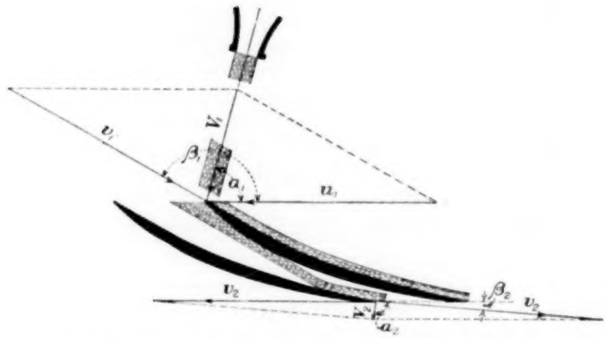


Fig. 6 See Continuation and Caption on Following Page



Peripheral Coefficient $\varphi = 2.00$: $V_1 = 0.98\sqrt{2gH}$: $A_1 = 75°$: $\beta = 151°$

(e)

Fig. 6 Diagrams Contrasting the Relative Velocities and Bucket Shapes of the Tangential Types of Impulse Wheels (a) and (b) with Similar Features of the Cross-Flow Type Wheel as Illustrated in (d) and (e). The Limit Between the Two Types is Shown at (c)

38 The cross-flow wheel does have an inherent disadvantage from an efficiency standpoint. The relative velocity between jet and bucket is the lowest (50 per cent of spouting velocity) in a tangential wheel and higher in the cross-flow types; for example, for a 100 per cent coefficient the relative velocity is practically twice what it is in a tangential wheel. This, of course, involves a greater friction loss, but in actual practice this may be offset by reduction of windage and splash losses, and particularly by the possibility of developing more power in a given working volume of space swept by the buckets than it is possible to do with the tangential type (see Fig. 5).

39 Vertical impulse wheels have certain mechanical advantages when large capacities are considered, these being similar to those which underlie the popularity and economies of the vertical reaction units, and in addition incorporate the possibility of feasible mechanical arrangement of more jets than possible with horizontal types. Engineers experienced with impulse work, however, are familiar with the tremendous disturbances that exist even in horizontal wheels having but a single jet, and can appreciate the condition that must exist when several jets are played on one wheel. When it is further considered that all the upwardly discharged water that falls or is spattered back into the path of the buckets must again leave the wheel at practically wheel velocity or half its own initial velocity, and that this represents a total loss, the desirability of getting the water positively and clearly away from the wheel is apparent. The cross-flow design, embodying as it does the feature of single direction of flow through the wheel without "backed" water losses, lends itself admirably to this arrange-

ment. This is particularly so as in extreme cases multiple or annular jets may be used without injurious interference, since each elemental jet uses such a relatively small portion (see Fig. 5) of the periphery as contrasted to the tangential type where each jet needs for its working space a chord subtending a relatively larger angle.

40 Fig. 7 presents the entire field of hydraulic-turbine practice. On this diagram are plotted the heads and horsepowers of practically every water wheel of note ever built, particularly those that extended the developed field in any direction, regardless of

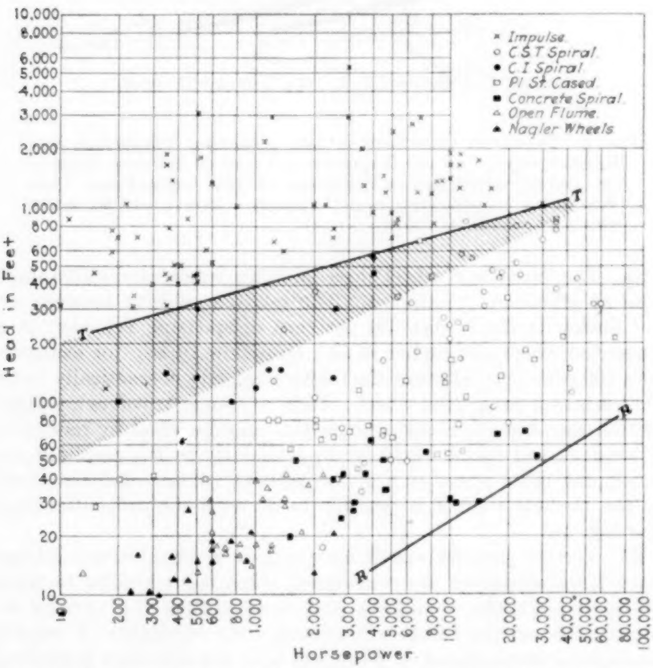


FIG. 7 A PLOTTING OF THE ENTIRE HYDRAULIC-TURBINE FIELD, EMBODYING PRACTICALLY ALL OF THE NOTEWORTHY PLANTS EVER BUILT

(Points above line *T-T* represent impulse units, practically all of them being of the tangential type. Line *R-R* represents approximately the limit of reaction units. The shaded portion adjacent to line *T-T* represents a field lying between the tangential impulse and high-head reaction fields and at present disadvantageously developed by them. This is the field to which the proposed cross-flow type of wheel is particularly adapted. Impulse-wheel horsepowers are plotted for one jet.)

type or nationality. The upper field above the line *T-T* (points marked ×) are of the impulse type. Those below the line *T-T* are of the reaction (Francis or suction) types. Portions below, to the right, or above the limits of the field of plotted points are as yet undeveloped.

41 Undesirably low generator speeds have so far restricted further development below and to the right of the line *R-R*. Size of parts and transportation limitations retard although do not prohibit extension to the right above the line *R-R*. The infrequency of points at the top of the sheet is due to the difficulties arising from extremely high pressures (above 3000 ft., for example), the difficulties in securing material for withstanding high stresses, and the comparative infrequency with which such high-head conditions are encountered. Undoubtedly the next few years will see units of 100,000 hp. capacity, and possibly more heads in the neighborhood of 3000 ft.; 70,000-hp. impulse units for 2150 ft. head are even now being worked out in detail.

42 The significance of Fig. 7 so far as this paper is concerned lies in the comparatively undeveloped gap existing between the impulse and reaction types. This gap exists by reason of the difference between the highest impulse characteristic speed (about 5 ft-lb. units) and the lowest reaction characteristic speed (about 10). Even the multiplication of jets and nozzles incorporated in most of the larger-capacity impulse units has failed to bring the two fields together.

43 The field of head and capacity now covered by impulse wheels of the tangential type needs extending only a limited amount. This extension is indicated roughly by the shaded section of Fig. 7, this section lying along and below the line *T-T*, between the fields to which tangential impulse and reaction wheels are well adapted. Going beyond this neutral ground in a lower direction involves competition with the reaction-type runners that give economical generator speeds with lower water velocities and better efficiencies. It is to this intermediate field that the cross-flow impulse design is peculiarly adapted, as with it characteristic speeds of 5 to 20 are readily obtainable with single jets, as shown in the upper shaded area of Fig. 2 and curve 14 of Fig. 4. With multiple or annular jets even higher speeds are possible, theoretically up to over 100 from curve 16, Fig. 4. There will probably be no commercial demand for such extreme speeds using the cross-flow principle, as the reaction type covers this field satisfactorily and with more desirable water velocities. The greater simplicity of the impulse type may, however, make those high-speed types work out advantageously with small or auxiliary units.

44 As in all simple developments, a detailed search of prior art shows numerous designs that might be construed to anticipate the cross-flow principle. These comprise various designs from the earliest forms of flutter wheel where water from a trough falls on a paddle wheel, up to and including various forms of Jonval wheels, which may have been set above tailwater, and through the various designs of Girard and Schwankrug wheels. All of these installations with which the author has been familiar have a flow other than tangential, not by reason of its advantage but because of certain desired mechanical arrangements and in spite of the acknowl-

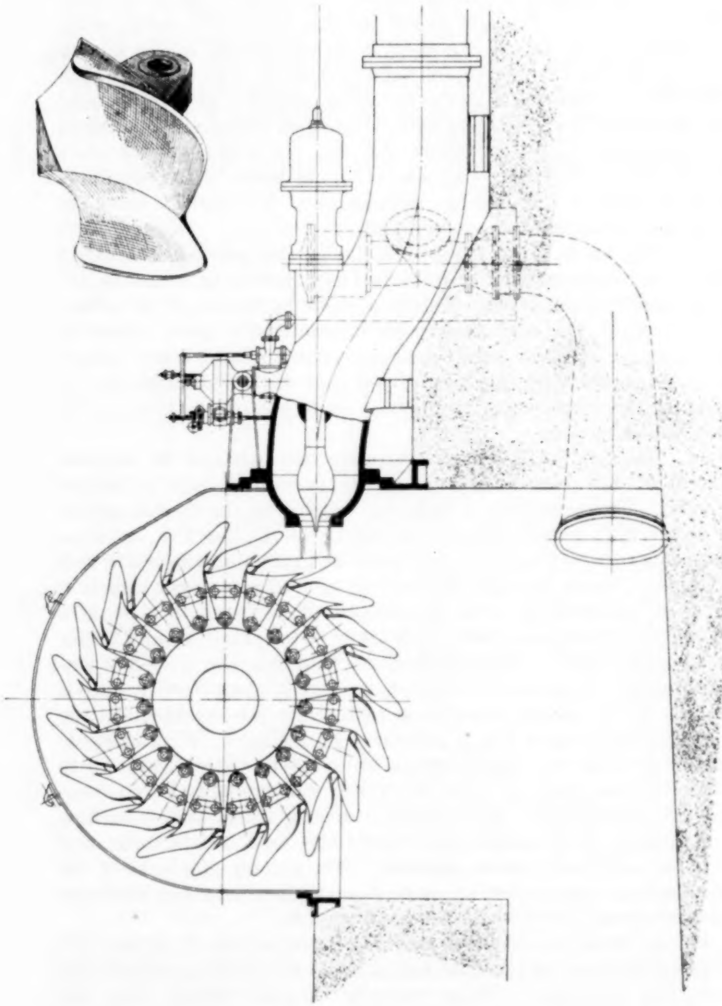


FIG. 8 SUGGESTED APPLICATION OF THE CROSS-FLOW PRINCIPLE TO AN IMPULSE-WHEEL UNIT

(See note at bottom of following page.)

edged disadvantage of departing from the tangential arrangement. This fact is shown by the numerous details of design which indicate the extreme measures that were used to secure flow as nearly as possible tangential. Exhaustive search has indicated no single instance where free circular jets, adjustable in size, have been purposely directed against the buckets at angles greater than 45 deg. with a tangent.

45 That the problem of securing higher specific speed in the impulse field is one of real commercial significance is indicated by the fact that such a variety of expedients have recently been used to raise speeds. These efforts parallel with surprising exactness the multiple-runner craze that dominated the reaction field during the years 1900 to 1910, resulting from the attempts of water-wheel builders to keep pace with electrical designers in speeds of units. The reaction program was confined at first to using multiple runners, as the entire periphery was used at the start. Later the reaction development followed lines of widening the inlet to the runner, increasing the velocity of the water, and finally in increasing the relative wheel velocity.

46 The impulse-wheel designer started with the limit of water velocity, has gone through the stages of multiple jets, then multiple wheels, and is now at the point of facing the use of larger percentages of wheel periphery (larger inlet area) and higher wheel speeds.

47 The analogy to the history of reaction-wheel development is perfect, even to the point where for a large field of head and capacity the electrical designer is ahead of the wheel designer in speed. With the development of high-speed impulse wheels there should be experienced a return to the single wheel and possibly to the vertical shaft setting, the two features that so completely dominated the tremendous advances made in the reaction field in the past decade.

48 This does not mean that the tangential wheel is to be displaced, but rather that it should be supplemented. For a certain range of head and capacity — a very wide range at that — it affords a perfect solution, one that harmonizes with desirable generator speeds, with excellent efficiency, and with the utmost simplicity and durability.

49 It is for lower heads and larger capacities that new designs are needed. While the elements of the machines themselves are not subjected to mechanical complication or hydraulic losses as serious as those which caused the multiple-runner reaction turbine to be superseded by the single vertical arrangement, the general complications of water passages and governing mechanisms and possibilities of flow interference are certainly such as to be expensive and undesirable. If the history of the progress in reaction

Note to Fig. 8. The large angle of incidence shown is obtained by simply raising the jet above a tangential line so that it flows in a more nearly radial direction. The sketch at the upper right of the figure shows a bucket for use with a cross-flow wheel and designed to illustrate dividing the discharge and equalizing the thrust by a splitter.

wheels is any guide, future developments in the impulse field will return to single runners of higher characteristic speeds just as the multiple-runner reaction wheels yielded first to higher-speed mixed-flow wheels and later to the still higher-speed axial-flow or suction wheels.

50 Fig. 8 is shown as an example of proportions and possible details of a unit based on the cross-flow principle for conditions of reduced head and increased capacity comparable with those which incur multiple jets or wheels. This is shown as indicating a possible trend in impulse design based on investigation of modern commercial needs and analysis of the hydraulic and mechanical features involved. It is not suggested that initial installations should be made for such large capacities and heads without first obtaining more operating data than are now available on the new type proposed. It is believed, however, that the low-head extension of the field of impulse wheels, particularly for small and moderate capacities, is warranted immediately.

51 The main purpose of this paper is not to present a complete solution of problems confronting the designer of impulse wheels, nor is it intended even to present the new type as being completely worked out. It is intended to draw attention forcibly to the possibility of departing from the orthodox tangential flow and 50 per cent coefficient, departing even from the specter of practice, precedent, usage, and textbooks without violating perfectly sound hydraulics. If the only too common tendency to get in a rut and stay there is lessened, the author will be more than satisfied.

DISCUSSION

William F. Uhl. Mr. Nagler in his paper before the Society in 1919 on the suction, or propeller, type of reaction turbine showed us the way to get higher speeds for low-head water-power units. He now shows us a way to get higher speeds with impulse turbines under such head and capacity conditions as are frequently encountered in the West.

Mr. Nagler has retained all the good features of the turbines as developed in recent years, simply changing the runners. Considered from a commercial standpoint this is very important, since it reduces the element of risk in adopting the new designs, as well as reducing the cost of new development work to a minimum.

Some of the most important things to consider in connection with hydraulic turbines are: mechanical simplicity, governing, pitting, efficiency, speed, weight, and runaway speed. Since no changes are so far contemplated in the design that will affect the first two features mentioned, we can assume that there will be no difficulties along these lines.

Regarding pitting and efficiency, we shall need further information before we can determine just where the new design will fit

in. The greatest difficulties are generally encountered when untried designs are used outside of the field for which they are suitable.

Efficiency is becoming more important with the greater use of mechanical power. One per cent increased efficiency will pay the fixed charges on a considerable investment where the output of the plant can be fully utilized. Increased speed in this case brings about a reduction in weight of the turbine and particularly of the generator, and in most cases will result in lower first cost of the power units.

The runaway speed of about 100 per cent for which the turbine runner and generator rotating parts must be designed where impulse turbines of the present type are used, adds a considerable item of first cost, and if this can be reduced it will probably be as large a factor in reducing the cost of impulse-turbine units as the increased normal speed due to the proposed design which Mr. Nagler has developed.

C. V. KERR. A comparison of the diagrams in Fig. 6 is instructive. The blades in (*b*) are shorter than those in (*e*), and due to the lower relative velocity will consume less energy in friction; the opening between the discharge edges is greater, and blade thickness will be less important; the discharge angles are larger, and less trouble would be expected in the jet leaving the wheel at that angle; the effect of windage will be to reduce the pressure between the blades in (*b*) and to increase it in (*e*). All these affect the ratio of final to initial energy in the jet, and hence affect the efficiency of the wheel.

Comparing the wheel in Fig. 1 with that in Fig. 8, it is obvious that the latter will have greater windage losses, especially at higher speed. If the location and direction of resultant pressure on the buckets could be determined, the relative moments about the shaft could be compared. Otherwise the advantage appears to be all in favor of the wheel in Fig. 1, as the jet is at the end of the vertical radius, while in Fig. 8 it is closer to the shaft and the leverage is reduced.

It appears from Fig. 7 that a number of these cross-flow wheels in the smaller powers and lower heads have been built. It would be gratifying if the author could supply performance data for some of them. So far they are in competition with the reaction rather than the tangential, type of wheel.

No. 1891

LIGNITE CHAR: ITS PRODUCTION AND POSSIBILITIES

By O. P. Hood,[1] Washington, D. C.
Member of the Society

Lignite char is lignite which has been dried and distilled in an oven especially designed for the purpose. About two and one-half tons of raw lignite reduce to one ton of char, the heating value of which is about 12,000 B.t.u. per lb. While raw lignite can be used satisfactorily in large steam-raising operations, the author believes that the search for a means to improve the fuel must continue. American lignites do not briquet well without the addition of a binder, and the Bureau of Mines has therefore been led to investigate the possibilities of an inexpensive carbonizing process and the use of the resulting lignite char direct without briquetting. This process is briefly described in the paper.

THE greatest difficulty with our lignite is the fact that in nearly every district where it should be the natural fuel it is put in competition with high-grade fuel. Canadian and North Dakota lignite must compete with anthracite and with Pittsburgh and Illinois bituminous coal; Texas lignite must compete with gas, oil, and Oklahoma bituminous coal. It is evident, however, that there must be a price at which the lower-grade fuel will begin to be attractive. In round numbers the ratio is somewhere in the neighborhood of half the price of good coal. With the rising price of bituminous coal we are fast approaching the time when this ratio will be common.

2 The handicaps of lignite are well known, but not always properly valued. The heating values of high-moisture fuels are somewhat misleading. The heat carried by the moisture is recovered and measured in the calorimeter, but is not fully utilized in a boiler furnace. The B.t.u. ratios, therefore, do not give the relative possible steaming values of the fuels if comparison is made between a high-moisture lignite and a low-moisture bituminous coal. Although the ash percentage may be low, there is usually a larger total amount of ash to handle in a plant using lignite. The fusing temperature of the ash is usually low, making high rates of combustion difficult and requiring larger grate areas and furnace

[1] Chief Mechanical Engineer, U. S. Bureau of Mines.

Contributed by the Fuels Division and presented at the Spring Meeting, Montreal, P. Q., Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers.

volumes than with higher-grade coal. Notwithstanding these handicaps, with present technique raw lignite can be used in large operations, and good efficiencies and reasonable capacities can be obtained. The problem is largely an economic one. When raw lignite is cheap enough in comparison with better coals it will be used in large steam-raising operations.

Improvement of Raw Lignite for Fuel Purposes

3 The search for a means to improve the fuel, however, must continue. A fuel classed as lignite in northern Bohemia, and weathering much as does our lignite, is as carefully prepared for market as is our anthracite. Seven prepared sizes are offered to the market. Raw lignite can probably be somewhat improved for steam raising by sizing the product more closely than is common practice. It is probable, however, that an improved lignite product must first cater to a special trade that will pay a special price. This is illustrated by the vision that has been so frequently held of improving the lignite by some process involving briquetting. Unlike the German "Braunkohle," our lignites do not make a stable and satisfactory briquet simply by drying the lignite and briquetting by heat and pressure. They lack sufficient inherent binder to consolidate and waterproof the mass. The necessary added binder increases the cost and hardly improves the quality. A quite satisfactory fuel can, however, be made by briquetting lignite char, and it is probable that some day such a fuel will be in common use.

4 There have been hopes that through the recovery of by-products sufficient credits might be obtained to lessen materially the cost of briquets. Profit can be shown on paper, but such a process is essentially a large-scale operation requiring a large investment and very substantial financial backing by those familiar with technical enterprise. It is difficult, therefore, to start such an industry, for there is no opportunity to begin small and grow up, returning profits into an improved plant. Capital familiar with technical enterprise finds less hazardous ventures, and capital unfamiliar with such enterprise is apt to be misled and lost.

Lignite Char and Its Possibilities

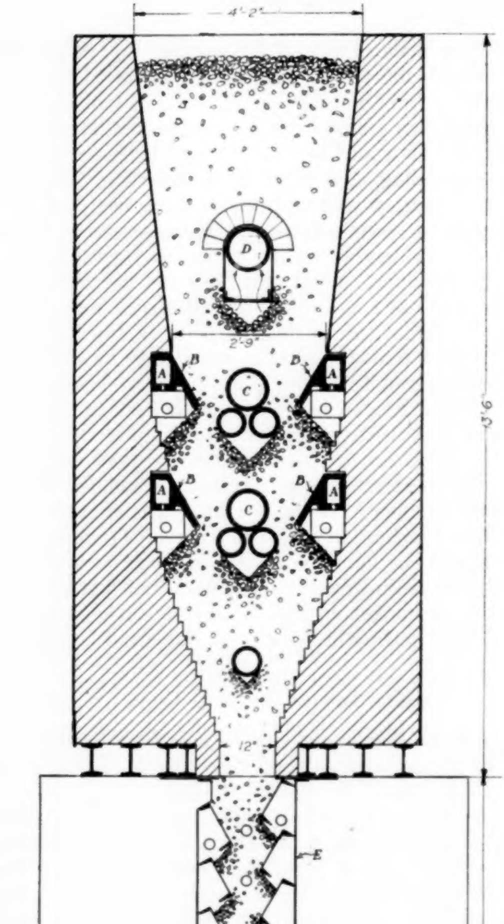
5 With these facts in mind, the United States Bureau of Mines is investigating the possibilities of a somewhat different program which has for its main features an inexpensive carbonizing device and the use of the lignite char direct, without briquetting. If a market for the char can be developed, and the small mine can produce char, there would be provided means for a natural evolution of an industry that in time might realize the larger vision of briquetting and recovery of by-products. Lignite char can best be described in a few words as a fuel rather near in analysis to anthracite,

but softer, with a little more volatile matter, and thus kindling easier. The volatile matter, in general, is from 8 to 12 per cent. The fixed carbon is from 65 to 70 per cent. The moisture is practically nil. Lignite char grades from pea coal to smaller sizes, and is a stable product. It would be easy to screen this material into two sizes corresponding closely to anthracite No. 1 and No. 2 buckwheat. Comparison with anthracite, however, does not give lignite char credit for certain favorable characteristics. With anthracite there is considerable carbon in the ash; with lignite char there is practically none. Char will hold the fire and burn out practically the same as a piece of charcoal buried in the ashes.

6 Whether a market can be developed for such a fuel at prices around five dollars a ton at the mine, remains to be shown, but it is at least encouraging to know that Germany used last year 400,000 tons of similar material for domestic heating and cooking. This fuel burns well with natural draft where a thin fuel bed, about 1½ in. in thickness, can be maintained. Base burners, cook stoves, and other heaters can be adapted to use the fuel satisfactorily. The Germans have developed a special stove, burning the fuel on a bed of ash in an enclosed drawer. There is no loss of fuel in the ash and our lignite char used in such a stove heats an oven sufficiently for baking operations and will boil water. It makes a very clean fire, is smokeless, and the char is clean to handle. It is, however, slow in getting under way as compared to a gas range.

Production of Lignite Char

7 To produce the char a very simple oven has been devised that greatly reduces the investment from that needed for ovens heretofore proposed. If lignite be passed through a combustion zone, moisture is first driven off; then combustible gases are distilled, and finally the solid carbon is burned. There is a considerable shrinkage in volume and a complete absence of caking quality. These steps are fairly distinct one from the other, so that the flow of lignite through the combustion zone may be so regulated that but little of the fixed carbon is burned. The combustion zone can be maintained by burning some of the distilled gases within the moving mass of lignite, and such direct heating is more efficient than where heat must be transmitted through refractory walls. The hot gases of combustion also pass through the mass, driving off the moisture and departing fairly cool. The furnace as finally developed by the Bureau of Mines, shown in Fig. 1, is something like an open-top lime kiln. The idea is to pass the lignite through two combustion zones. While three or four such zones might be used, two appear to be satisfactory. Combustion is maintained by air from the four ducts *A* in the side walls. The flow of lignite is most rapid at the hottest zones between the central circular baffles *C* and the inclined side baffles *B*. Gases are exhausted from the upper perforated outlet *D*. The char is continuously dis-



Cross Section

FIG. 1 BUREAU OF MINES OVEN FOR PRODUCING LIGNITE CHAR

charged after passing an enclosed cooling device *E*, in which it is protected from the air. The cost of this oven is about $150 per ton of capacity. The process has proved simple and efficient. Of the gas driven off, much of it is used in the combustion zone, and in addition, less than 5 per cent of the weight of the original lignite is burned. That is to say, the fixed-carbon loss in the process for drying and distilling is lower than is usually found for drying alone where separate driers are used. The char obtained by such a process may, of course, be briquetted.

8 An oven of this sort was operated at Grand Forks, North Dakota, during the past summer, and about 400 tons of various North Dakota lignites passed through. In February about 100 tons of Saskatchewan lignite was tried to discover whether this presented any special problems.

9 About two and a half tons of raw lignite reduce to one ton of char, and the heating value is about 12,000 B.t.u. per lb. The moisture is very low, and the char can be stored without danger of fire or degradation in size. Where the freight charge is heavy, it would be an advantage to ship char instead of raw lignite.

DISCUSSION

Max Toltz. According to the United States Geological Survey, 800 billion tons of lignite are deposited in North and South Dakota and Montana. While little used heretofore, probably due to the low price of eastern coal and the high freight rates on lignite as compared with rates from the Illinois coal fields and Great Lakes docks, the Northwest is now realizing that lignite is the rational fuel for that section of the country. While some steps have been taken toward its utilization, certain difficulties must be overcome before lignite can compete with coal and other fuels.

In general, lignite has been considered but little better than peat for heat generation, and has been used only by people in the neighborhood of the deposits. About eight or ten years ago an attempt was made to briquet lignite, but it was not found practical to do so without a binder.

The disadvantages of lignite as compared with bituminous coal are its high moisture content (30 to 35 per cent) and behavior when weathering, with the resulting shrinking, cracking, and disintegrating, usually called "slacking." Before attempting to establish the possibilities of using northwestern lignite, we may consider the means and facilities evolved by the Germans since the close of the World War.

Germany's iron-ore holdings were by the peace treaty reduced from $25\frac{1}{2}$ per cent of the total ore in Europe to $7\frac{1}{2}$ per cent, and her coal resources from 40 to 30 per cent of the total. In addition Germany was required to deliver to France 25 million tons per year of high-grade coal from the Ruhr and two million gallons of benzol.

It was therefore necessary to develop the lower grades of coal, such as lignite and brown coal, for domestic and industrial consumption. It was also essential to recover the by-products from these low-grade fuels, especially the oils. The first step was to use these low-grade fuels directly under boilers as fuel. Central power plants were built adjoining the lignite deposits so that the fuel as mined could be delivered directly to the boilers without storage, which would tend to disintegrate or slack it. In one of these power plants, distributing power over a radius of 120 miles, in the majority of the boilers step grates were used on which the lignite was burned successfully. Chain-grate stokers were also used, but care had to be taken to prevent the finer lignite from sifting into the ashpit. This station burned 900 tons of lignite per 24 hours.

Another successful method of burning lignite under boilers was by means of a combination of semi-gas producer and step grates. The furnace is a Dutch oven, in which a chamber of a gas producer is partitioned off from the lower step grates by a wall and arch through which the gases escape to the combustion chamber over the step grates. The incompletely consumed lignite from the producer falls through its bottom upon the step grates, where it is finally burned. The gases of combustion from the fuel on the step grates combine with the off-going gases from the producer and are ignited by an arch back of the grates. The process in the producer is low carbonizing of the fuel, so that the lignite falling on the step grates is practically half coke. The air for combustion is supplied partly by fans under the step grates and partly at a point under the back arch, after being heated in ducts in the setting. This latter air mixes with the gases of combustion. The following results have been obtained in this type of furnace and boiler installation with lignite of 12,060 B.t.u. containing 43 per cent of moisture and 6 per cent of ash:

Evaporation from and at 212 deg. fahr.	6.88 lb.
Efficiency of furnace and boiler	65.75 per cent
Chimney losses	21 per cent
Losses due to ashes, radiation, etc.	12.5 per cent
CO_2	13.5 per cent

When using a portion of the chimney gases to pre-dry the lignite, an efficiency of from 75 to 80 per cent has been obtained, it is claimed. Lignite was burned at a rate of from 60 to 100 lb. per hr. per sq. ft. of grate, and peat at rates up to 130 lb.

In these processes of combustion no by-products are recovered. To recover by-products and to produce oils of all kinds and benzol, the method of carbonizing the lignite is extensively used. The lignite is coked, the coke being finely briquetted. The condensable gases are distilled and the oils and other by-products recovered. The non-condensable gases are burned in the retorts and ovens and the off-going heat of combustion pre-drys the lignite to be coked. It was especially stated by one of the engineers that this process

of by-product recovery would hardly pay in the United States because of the relative cheapness of oils here.

In view of the progress made in Germany in developing methods of using lignite and brown coal, it is time that in this country ways and means were found to make available, at least for the Northwest, the immense deposits of lignite, not only to conserve our fuel resources but also to relieve the transportation system of tonnage hauling which could be used advantageously otherwise. The author has shown at least one way of accomplishing results by making lignite char. I would suggest improving this process by using the off-going heat to pre-dry the lignite. Although this process has not yet been used on a commercial scale, there is no doubt that it will be economical. The author's suggestion as to prepared sizes of lignites is good, but we should go farther. If the lignite is to be shipped to distant points it should have not over ten per cent moisture — to cut down freight charges on the combustible. The mechanical drying of lignite should not be excessive in cost. Even if the lignite shrinks or slacks, good results can be obtained by burning it on proper grates, such as step grates, or in a combination gas producer and step grate, one such installation being in use at Williston, North Dakota. The stack gases should be used to further dry out the lignite. Lignite may be dried and pulverized at the source, and so shipped in closed cars, although it may cake because of the long-distance haul.

It has been demonstrated in a laboratory test that North Dakota lignite can be briquetted without a binder after the sulphur has been extracted. No further details can be given at this time, but this process may lead to a method of briquetting which, if successful, will solve the domestic fuel problem of the Northwest. During the winter of 1922–1923 about 5000 tons of North Dakota lignite were used in the Twin Cities, Minneapolis and St. Paul, Minn., for domestic fuel under special instructions issued by the mine owners. The users have been generally satisfied with this fuel, although there has been complaint of the odor accompanying combustion.

It is gratifying to report that in addition to the United States Bureau of Mines, the Mining Department of the University of North Dakota and the government of the province of Saskatchewan are working for better methods of burning lignite.

Lesslie R. Thomson.[1] The Lignite Utilization Board of Canada is a body created by Order in Council of the Dominion Government. The Board represents the Federal Government, the government of the province of Manitoba, and the government of the province of Saskatchewan. It was created in 1918, on the recommendation of the Research Council of Canada, to demonstrate the

[1] Lignite Utilization Board of Canada, Montreal, Canada.

commercial possibilities of a process of providing a high-grade domestic fuel to compete with American imported anthracite in the Central Canadian West. We are mainly concerned with the production of a high-grade domestic fuel, and do not touch the power question. It is quite possible today to burn lignite satisfactorily in power installations, as already pointed out. The handicaps of raw lignite have already been touched on, the slack and the moisture, but there is one point in connection with lignite that some overlook, especially in connection with the briquetting or transshipment of raw lignite. It is hygroscopic. You may dry the lignite to any degree, but if exposed to air it reabsorbs from 13 to 16 per cent moisture unless shipped in sealed containers. This makes it inexpedient to briquet it raw unless a sufficient amount of binder is used to waterproof it against that absorption.

As to by-products, this whole process, so far as the Canadian and probably the American Northwest are concerned, is not feasible at present, because it is certainly uneconomical to manufacture by-products at high cost unless they can be marketed. There are no markets in the Canadian West for by-products (except perhaps motor fuel), and until such time as farmers are demanding fertilizers it is not economical to produce them. The only by-product that is likely to have an immediate market is a motor fuel for Ford cars and tractors.

Mr. Hood's paper has looked on lignite char as a product of itself to be marketed independently, or as a finished product. We look on it as an essential stage in the general process of producing a high-grade briquet. Our first effort was directed to that end, and as we found that there were no commercial carbonizers or retorts on the market that would be suitable for the retorting of the raw lignite, we thereupon developed one. We first built a small retort, consisting essentially of a floor at an angle of 45 deg., which by experiment had been found to work fairly satisfactorily. The lignite slid down this inclined floor with heat underneath, and the thickness of the stream of lignite was regulated by vertical baffles which had the effect of making the particles tumble over each other. This type of retort would give by-products, when and if necessary. We built one about 20 in. long, which at first refused to work. We eventually built one of semi-commercial size, 10 ft. on the floor, which also at first refused to work. Ultimately, however, both were operated very successfully. From these two we got many data on the possibilities and capacities of this type of retort. With these data we felt justified in building six full-sized retorts, 20 ft. on the floor. These also refused to operate. We made certain changes and we got them to operate haltingly, but the operation could not be termed commercial. It was too expensive, and we had troubles due to bad floor material, unsatisfactory refractories, and other items. However, the principle works but probably cannot be made commercial.

We are now erecting one of the carbonizers as indicated in Fig. 1

of the author's paper. We are attempting from the beginning to draw off the small excess of gas, not with the idea of using it for the moment, but of improving the operation of the retort itself. We do not regard it as having reached a complete commercial stage as yet, and we wish to give it a further trial before embarking on a larger installation.

A word or two in regard to technical points may be in order. A series of laboratory experiments showed that the B.t.u. content of the char varies in a certain way with the temperature of carbonizing. The curve reached the peak at about 575 deg. cent. This degree of carbonizing corresponded approximately with 10 to 11 per cent of volatile matter retained.

The next point is that the char is not salable at all as far as we can see at present; but if a high-grade briquet be made of it, then that briquet becomes salable immediately, because all furnaces in the Canadian Northwest are designed to burn American anthracite, and it is primarily with that fuel that we shall compete. Since the inception of the Board we have come to the conclusion that we must also compete with our better-grade Alberta coals, but I believe a high-grade briquet can be made which will compete with both and give excellent service for domestic heating. Therefore all discussions on the use of lignite char in the Canadian Northwest in power installations are, for the present, a little beside the point.

The next point is the appearance of the char. Our screen analysis corresponds very closely to that given by Mr. Hood. All our char will pass through a one-eighth mesh and be retained on about 170 or 180 mesh if it is retorted at a fairly high temperature. Each particle from the Grand Forks retort has a silvery sheen. In our own retorts we do not get that sheen, due probably to the fact that as the gases come off they are immediately taken away and the carbonized lignite itself as it moves down is not exposed to any high temperature. On the other hand, with the retort of Dr. Babcock the gases come up and are above lignite particles. It is believed that the silvery sheen is due to deposits of carbonized lignite tar on the particles. It is not known whether or not the silvery sheen char makes a better material for briquetting. On the whole, however, the problems of briquetting are fairly well solved, certainly in a laboratory way, and it only remains to solve the ordinary commercial layouts.

Two criticisms for the future of the retort shown in the paper may be made. First, it must be fed certain screened sizes. In a country where a large amount of lignite slack and small sizes is for sale, it would seem expedient when attempting to make a high-grade fuel that raw fuel of the cheapest material should be bought, and if the retort will not use it, a very large proportion of the possible supply is limited. The second criticism is on the fact that although there is provision for a small amount of by-product recovery, the final solution of our lignite retorting will probably be

found, especially as the population increases, in a retort that will give the complete by-products if and when necessary.

E. N. Trump. I was much impressed in Germany in 1913 with the performance of a new type of furnace for burning lignite, which was on the order of a rotary furnace. It was a cylinder about 12 ft. long, lined with refractory firebrick, and fitted with a pick-up device, at the end nearest the furnace, consisting of a scoop which took in a certain amount of the material at every revolution. At the end away from the furnace was a hole through which the air for combustion entered. That furnace gave a long flame of very high temperature. The lignite entering the furnace was first dried by the radiated heat from the brick, and then as it became char it burned by the direct entrance of the air, which was regulated, at the other end. The air was not admitted in sufficient quantities to burn the fuel entirely, but it made a gas that was carried on into the furnace and the air for combustion was introduced around the side of the revolving furnace. Much better efficiency was obtained than with the step grate, and for boiler use it should be almost ideal. While we have not heard of a rotary furnace for making char, it should be quite satisfactory if the lignite was heated at low temperatures with a very small amount of air.

H. D. Savage. From the little experience we have had, burning lignite in powdered form presents the simplest and most efficient way of accomplishing the end desired. Some years ago, in Oklahoma, about seven carloads were burned by Morris & Company in pulverized form. No change was made in the plant. In the tests an efficiency of about 78 per cent was secured in that plant on this lignite at the average capacities demanded. More recently, at Lakeside, about five carloads of Colorado brown coal, which is a little better grade than lignite, were burned. No change was made in the plant. One boiler was set aside and the fuel was put through the pulverizing plant and burned in regular operation. The boiler operated at from 200 to 250 per cent of capacity.

Drying of the lignite does not seem to be necessary. The moisture content of the lignite burned at Lakeside was about 22 per cent. The lignite lost a great deal of moisture by air drying, at least five per cent the first day, and this loss gradually decreased to about one per cent on the last day of the test. A plant is now being built at Boulder to utilize this coal. It would seem to have great possibilities.

In burning any coal in pulverized form, with either high ash or sulphur content, the difficulties that are met with under ordinary methods of burning are not present. In South America coal is being burned on fourteen locomotives that contains at least 30 per cent ash and from 7 to 14 per cent sulphur. No difficulties are experienced with it except that it turns the coaches white. The

Brazilians have spent a large sum of money to develop some method of using their native coal, and powdered fuel seems to be the answer. Of course the sulphur itself has some heating value, about 4000 B.t.u. per lb. and by pulverizing this is brought out.

THE AUTHOR. We have assumed lately that for steam-making purposes there is no need of processing lignite. It has been shown that powdering the fuel is a good plan. It is known that underfeed stokers will handle lignite in a satisfactory manner. In a test in Fort Worth, Texas, on a chain-grate stoker at 275 per cent of boiler rating, the efficiencies quoted are comparable with the efficiencies obtained in using good bituminous coal, i.e., efficiencies above 70 per cent, so that that phase of the problem is not interesting us at the present time.

Answering Mr. Toltz, the trial carbonizing device shown does use the heat in the gases, but the gases leave the device at a low temperature. In fact, that was one of the troubles in operating. The gases went off at such a low temperature that they did not burn, and the smell was such that we had to raise the temperature of the gases in order to burn it. In regard to reducing the sulphur content of lignite to obtain a product which would briquet, this should be compared with a statement that comes from Oregon. There, with their lignite, by the addition of sulphur, they are producing briquets.

The problem of raising steam, we feel, is reasonably solved with present-day devices. In Germany the step grate is almost the universal practice.

The distillation of brown coal is very interesting. A bed of brown coal as it lies in the ground in Central Germany has a banded formation, varying all the way from a cream color to a barely distinguishable lighter shade than the black brown of the main body. These special veins or beds are used for distillation. It affects the mining problem. Instead of mining the whole mass, these beds are mined by hand and kept separate from the rest. Some of that will run as high as 13 per cent of paraffin. Since the moisture content is about 50 per cent, the percentage of paraffin on a dry basis is very high. That paraffin is the base of a considerable string of products, some of which are the lubricating oils. So, when we speak of the German background to this industry, we have to differentiate. They are not taking the whole lignite, but are taking particular parts. We have our own material and problem and must solve it in our own way and not merely import a way from abroad.

No. 1892

RECENT DEVELOPMENTS IN BALANCING MACHINES

By Carl Richard Söderberg,[1] East Pittsburgh, Pa.

Non-Member

In this paper the author describes a machine for balancing quickly and at low cost light rotating masses such as the rotors of small electric motors. The problems involved are discussed and the solution presented and illustrated. Four appendices are devoted to the discussion of (a) the correction of a general state of unbalance by two masses, (b) the requirements for constant period of a balancing machine with a movable fulcrum, (c) the analysis of the vibrating motion, and (d) the sensitiveness of the balancing machine.

THE balancing of rotating machinery is still far from being an exact science, although it has received a great deal of attention during recent years. Certain aspects of the problem are still giving difficulties. The balancing of small high-speed motors for domestic use has been neglected to a great extent because satisfactory balancing equipment has not been available. The necessity for balancing this type of rotating apparatus is continually growing with the increase in speed and capacity and the customers' demands for a lasting product. The present paper is a brief description of the efforts made to solve this pressing problem and of the balancing machine finally developed that eliminated the difficulties and made it possible to manufacture quietly running motors.

2 Several fundamental difficulties, in addition to those encountered in balancing rotors of ordinary size, are involved in the problem of balancing small rotors.

a The balancing operation must be performed in a few minutes because the total cost of each unit is so low

b The high speed necessitates a very accurate balance

c The actual values of the centrifugal forces that are produced by a considerable unbalance are so low that it has been found extremely difficult to obtain an indicating device for the vibrating motion.

3 Because of these difficulties several attempts to produce a balancing machine for small armatures, operating on the usual

[1] Engineer Motor Engrg. Dept., Westinghouse Elec. & Mfg. Co.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers.

principles of separating the static and the dynamic unbalance, have failed. It was decided, therefore, at the beginning of the investigation to follow a plan that would procure complete balancing in one operation, without previous static balance. Arrangements of this type have been used before, but in these the balance is obtained by three weights, two of which cause considerable inconvenience in computing.

4 It is shown in Appendix No. 1 that the general state of unbalance can be corrected by two masses, one in each of two transverse planes of the body. This can easily be understood by recalling the fact that any system of forces emanating from a straight line may be counteracted by two forces.

5 The most common type of balancing machine now on the market consists of a vibrating table so supported on springs and a fulcrum that the table is capable of performing small vibrations around a fixed axis. On this vibrating member are mounted the rotor to be balanced, a "dynamical image" of this rotor, usually consisting of a variable couple, and a driving mechanism for rotating the body and its image. If the fulcrum axis is parallel with the axis of rotation the unbalanced dynamic couple is eliminated, and the static unbalance can be ascertained and corrected. By shifting the fulcrum axis to a position at right angles to the axis of rotation, the remaining couple will be effective and may be determined and corrected. In both instances the vibrating table performs a forced vibration under the influence of the unbalanced forces in the body, and the amplitude of this motion is increased by adjusting the speed of rotation so that resonance occurs. Thus the sensitiveness of the machine is magnified without undue increase of speed. The original dynamic balancing machine, the squirrel-cage machine devised by Akimoff[1] several years ago was the first arrangement of this kind. Quite frequently the static balance is performed on parallel ways.

6 The static-dynamic balancing machine has the following fundamental disadvantages:

a The static balancing generally increases the magnitude of the unbalanced couple because of incorrect longitudinal location of the correction weights. Thus the amount of balancing weights is greater than necessary

b The number of places in which correction weights are applied is frequently four and cannot be reduced to less than three, while the theoretical minimum is two

c Any error in the static balancing will have a serious effect on the determination of the couple. The effect of residuary static unbalance is frequently magnified on account of the large moment arm. In order to obtain a balancing result with a certain tolerance it is therefore

[1] Dynamic Balance, by N. W. Akimoff, Trans. A.S.M.E., vol. 38 (1916), p. 367.

necessary to perform the static balancing with exaggerated accuracy.

7 With the disadvantages in the existing equipment and the special requirements for the balancing of small rotors, the following demands were made on the new type of machine:

a Balance should be obtained by adding the theoretical minimum of weights to the body; that is, the two theoretical masses necessary for counteracting the system of unbalanced centrifugal forces should be given in the result

b The balancing operation should be performed with uniform accuracy; that is, the effect of one mass should be eliminated while the other mass is determined

c The balancing operation should take but a short time and be inexpensive, so that it can be applied to quantity production.

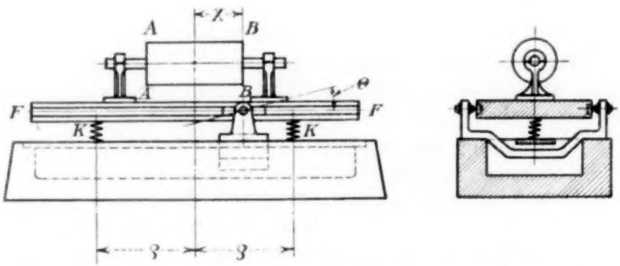


FIG. 1 SCHEMATIC ARRANGEMENT OF BALANCING MACHINE

BALANCING MACHINE WITH MOVABLE FULCRUM

8 Fig. 1 shows schematically an arrangement whereby the general state of unbalance may be obtained without segregation into static and dynamic unbalance. The body to be balanced is mounted in bearing blocks on a vibrating table supported by two spring members and a movable fulcrum member. This admits of motion around any axis in the plane *F-F* at right angles to the axis of rotation. Assuming that the transverse planes *A-A* and *B-B* have suitable arrangements for the application of correction weights, it is certain, from the above, that any unbalance in the body may be corrected by one weight in each of these planes. By placing the fulcrum axis in one of these planes, say, *B-B*, the theoretical weight in this plane is eliminated as far as its effect upon the motion of the vibrating table is concerned. This will now be produced by the unbalance in the other plane.

9 If a counterbalancing device for measuring the amount and the angular position of this weight is added to the arrangement, the unbalance may be determined in the same manner as static

or dynamic unbalance is determined in the old type of balancing machine. By subsequently placing the fulcrum in the plane *A-A* the effect of the unbalance in this plane is eliminated, and the same procedure may be applied for determining the weight in the plane *B-B*. It is evidently of no importance whether the balancing weights are applied during the operation or not, since the effect upon the vibrating table is reduced in each case to that of one unknown weight. It is also evident that this arrangement meets the requirements.

10 This arrangement, which has been called a balancing machine with a movable fulcrum, was adopted as the most suitable one for balancing small armatures. There is, however, no indication that its application will be restricted to small rotors.

Features of the First Design

11 The subsequent design of an operative machine offered a number of interesting problems. There are two possibilities with reference to the operation of the device:

a Determining the magnitude and the location of the unbalanced masses by means of a counteracting device

b Determining the magnitude of the unbalanced masses by the amplitude of the motion, and the location by "cut and trial" or by "marking" the rotor.

12 Although these two systems are different, they have no bearing upon the following theories. The first model was built and operated in accordance with the second alternative mainly because unbalance on the type of motors for which the model was built could easily be corrected by inserting wedges into the slots. It seems probable that, in this particular case, no gain could be obtained in speed of operation by applying a counterbalancing device. The following principles relate to either one of these alternatives and will be discussed separately for the sake of clearness.

13 *Speed Variation for Different Locations of the Fulcrum Axis.* Since the moment of inertia of the vibrating system varies with the location of the fulcrum axis, it is to be expected that the natural period, and hence the resonance speed, should vary. It has been found, however, that an arrangement can be made in which the restoring element receives a corresponding variation, so that the quotient remains constant. The complete theoretical explanation is given in Appendix No. 2. The result of this investigation shows that if the two spring members are identical and located symmetrically with reference to the center of gravity of the vibrating bed and at a given distance, the natural period of the system is independent of the fulcrum location. Fig. 2 gives an illustration of the principle for the specific case where the center of gravity of the vibrating system is located in the fulcrum plane.

14 A constant balancing speed is very essential for efficiency and accuracy. The latter is largely dependent on the degree of resonance, and a slight error in the speed may distort the result altogether. The type of indicating device that was adopted served to increase the importance of constant speed.

15 Naturally the principle does not admit a variety of armatures to be balanced on the same set-up. As far as small rotors are concerned, they are manufactured in large lots, thus making it necessary to have a balancing machine for each type. When frequent changes have to be made, a vibrating table may easily be supplied for each type of armature.

16 *Indicating Device.* The indicating device proved to be a stumbling block. Attempts to use the ordinary type of dial gage indicators failed altogether. There was always the possibility of using the conventional mirror and light beam, but it was not considered practical in production routine. A vibrating reed was finally applied with success. Spring steel 0.125 in. by 0.006 in. is

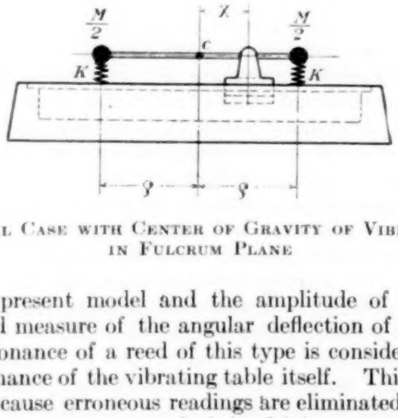


FIG. 2 SPECIAL CASE WITH CENTER OF GRAVITY OF VIBRATING SYSTEM IN FULCRUM PLANE

used on the present model and the amplitude of the reed is a tolerably good measure of the angular deflection of the vibrating bed. The resonance of a reed of this type is considerably sharper than the resonance of the vibrating table itself. This is a decided advantage, because erroneous readings are eliminated. It necessitates, however, a constant speed of the driving arrangement.

17 *Driving Mechanism.* The requirements for constant speed magnified the importance of the driving motor. Several attempts were made with a small motor mounted on the vibrating table, but without success. The kinetic energy of the rotating masses must be sufficient to resist variation in the friction load. As long as the driving motor is mounted on the vibrating table, its weight must be restricted in order that the vibrating mass may not be too great. In the arrangement used at present the rotor is being driven by a fairly large a.c. motor mounted on solid foundations. The driving belt consists of a silk ribbon. This arrangement has proved to be satisfactory. The disturbance of the belt is not sufficient to influence the motion of the vibrating bed.

18 *Predetermination of Sensitiveness.* The problem of pre-

determining the amplitude for a given unbalance necessitates a closer study of the factors that limit the motion at resonance, and a better knowledge of the internal friction of springs is therefore of extreme importance. Little work has been done along this line, however, and there are no reliable methods available for the predetermination of the internal friction in springs. It is generally assumed that the damping of a spring is a linear function of the speed, and this appears to be fairly true for small motions. A variation of the spring characteristic for different degrees of compression will also limit the amplitude at resonance, although in a manner different from that of internal friction. If this variation is large it may seriously impair the resonance. The requirements of springs for balancing machines are therefore:

a Straight-line characteristics
b Low internal friction.

19 The relation of the amplitude, that is, the angular deflection of the vibrating bed, to the unbalanced moment is explained in Appendix No. 3. It is shown that the angular deflection is directly proportional to the unbalanced moment with regard to the fulcrum axis and inversely proportional to the damping factor. This is true for perfect resonance and for the assumption that the damping is proportional to the first power of the speed. It has also been found in this investigation that the major part of the damping effect occurs in the springs themselves. The total damping is therefore a function of the location of the fulcrum axis. The minimum of damping occurs at the central location of the fulcrum. Therefore the amplitude is not a direct measure of the unbalanced moment; it is always possible, however, to place the rotor in such a manner that the readings have the same value for the two planes in which the unbalance is corrected, because the variation of the damping factor is symmetrical with reference to the central location of the fulcrum.

20 There is, however, another factor of importance in this connection. It is theoretically true that if the springs have straight-line characteristics and a sufficiently low internal friction, a small disturbance (unbalance) will produce a considerable amplitude. If the disturbance is small, however, the time required by the vibrating system to absorb sufficient kinetic energy to attain its maximum amplitude will increase. This introduces another element into the problem, namely, the proportion between the moment of inertia of the vibrating system and the magnitude of the disturbance. Common sense tells us that there is a certain limit to the moment of inertia of the vibrating system for each amount of unbalance to be measured, but no attempt has previously been made to establish actual values. In the author's opinion, the time required for the balancing machine to "pick up" and the amount of internal friction are the two elements that enter into the question of sensitiveness. It is shown in Appendix

No. 4 that this time element is largely dependent on the quantity P/I; that is, the proportion between the unbalanced moment and the moment of inertia of the vibrating system. The minimum value of P/I will thus be a logical measure of the sensitiveness of the balancing machine. Below are given minimum values of P/I that were obtained in the model with a movable fulcrum and three other balancing machines of the usual type.

		Value of $(P/I)_{min.}$
1	Model with movable fulcrum	15×10^{-6}
2	125–500-lb. balancing machine	0.6×10^{-6}
3	500–3000-lb. balancing machine	0.45×10^{-6}
4	3000–10,000-lb. balancing machine	0.4×10^{-6}

21 These values were obtained by experiments on the balancing machines. They are influenced, therefore, by the following factors:

a The nature of the indicating device
b The damping factor
c The actual value of the moment of inertia of the vibrating system with regard to the axis of oscillation.

22 The model with a movable fulcrum has an indicating device which records the angular deflection of the vibrating bed so that only the sensitiveness of this device (amplitude per radian) enters into the expression P/I. The other machines, however, are equipped with dial gage indicators so that in the sensitiveness of the indicator is also included its distance from the axis of oscillation. This distance is usually made as large as possible. The larger machines therefore have the advantage of a more sensitive indicator. The dial gage indicators on these machines are all of the same type. A total amplitude of 2 deg. (0.002 in. linear motion) is considered the minimum detectable motion.

23 The damping factor and the moment of inertia are variable in the model with a movable fulcrum. Therefore the value of P/I refers to the central location of the fulcrum.

24 The balancing machines in cases 2, 3, and 4 are of the same type. They show a tolerable consistency in the minimum value of P/I with a slight increase for the lower capacities. The model with a movable fulcrum has a capacity of an entirely different order and the value of P/I is increased correspondingly. This illustrates the difficulty in obtaining an accurate balance of small rotors. In order to keep the minimum unbalance sufficiently low it is necessary to have the moment of inertia of the vibrating system exceedingly small.

25 It would be highly desirable if the limiting values of P/I could be learned when a balancing machine is purchased. The buyer would then be able to tell if the lower limit is consistent with the accuracy of balancing that he wishes to obtain on the smallest rotor to be balanced. At the present time it is the practice to rate balancing machines with reference to the maximum and minimum weights of rotors to be balanced. The upper limit

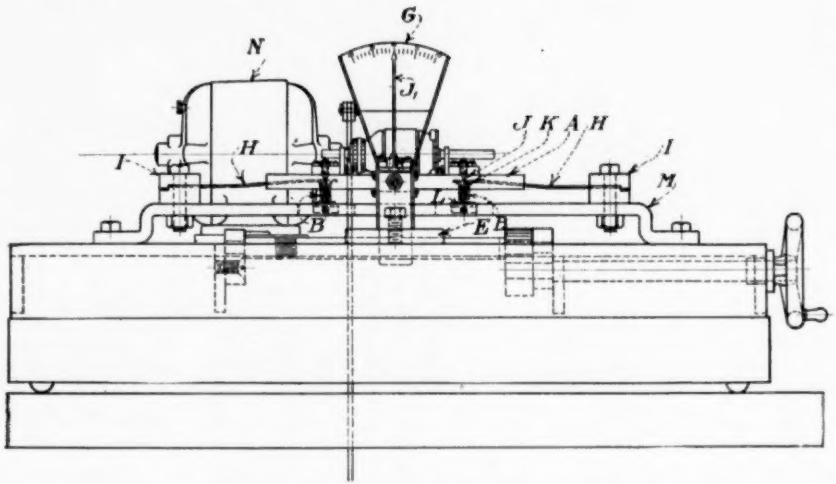


FIG. 3 SIDE VIEW OF BALANCING MACHINE WITH MOVABLE FULCRUM

may be necessary from the point of view of stresses in the mechanical structure. The lower limit does not, however, give reliable information with regard to the accuracy of balancing.

DESCRIPTION OF FIRST MODEL

26 Figs. 3, 4, and 5 show a balancing machine with movable fulcrum adapted to balance an electric armature weighing about $9\frac{1}{2}$ oz. Fig. 3 is a side view, Fig. 4 an end view, and Fig. 5 a horizontal view. The machine is shown mounted on a working bench with a foot-operated starting and braking arrangement of the driving motor.

27 The vibrating bed *A* is supported by the coil springs *B* and the pivots *C*. These pivots engage in centers on the sliding blocks *D* so that by moving the carriage *E* the axis of oscillation is moved relatively to the armature. The location of the pivots with reference to the armature is indicated by the pointer *S*. One of the sliding blocks carries the vibrating reed *F*, which vibrates across the graduation on the scale *G* attached to the carriage *E*. The indications of the reed are therefore proportional to the angular deflection of the vibrating bed.

28 The bed is prevented from moving in the horizontal plane by the flat springs *H* clamped in the blocks *I*.

These flat springs press against the knife edges J which have center pins engaging in centers on the caps K of the coil springs. The flat springs have slots to fit these pins. The slots are machined so that there is practically no clearance in the lateral direction but a considerable clearance in the longitudinal.

29 The coil springs are supported on the longitudinal member M by the adjustable seats L. Thus, the length of the springs may be adjusted so that the entire weight of the vibrating system is carried

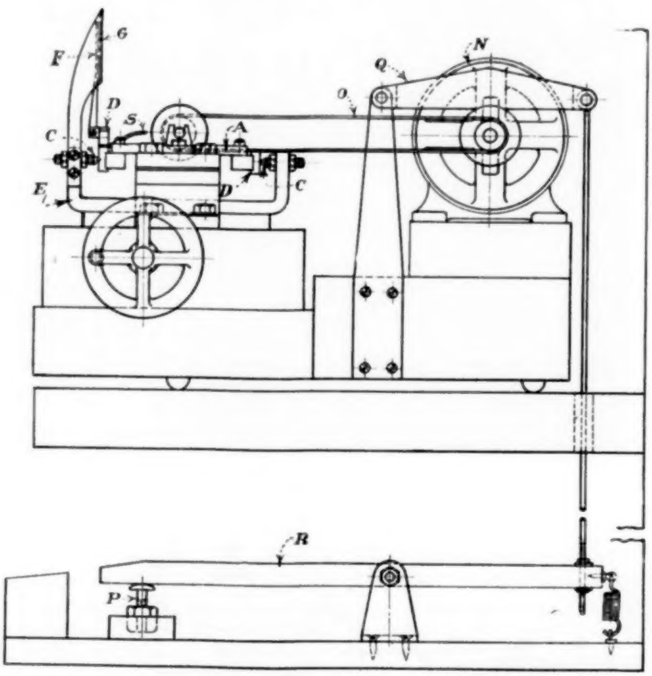


Fig. 4 End View of Balancing Machine with Movable Fulcrum

by the springs. In this case the pivots carry no other load than that of the inertia forces. The carriage E moves underneath the spring-supporting member M. This admits of moving the fulcrum member past the springs if desired.

30 The driving motor N is a single-phase induction motor starting as a repulsion motor. The load is very small so that it runs at its synchronous speed. The armature to be balanced is driven by the belt O. The critical speed of the apparatus is about 1700 r.p.m. The operating speed of the armature is 10,000 r.p.m.

31 The operation of this balancing machine is as follows: The armature is mounted in the machine and the carriage E is

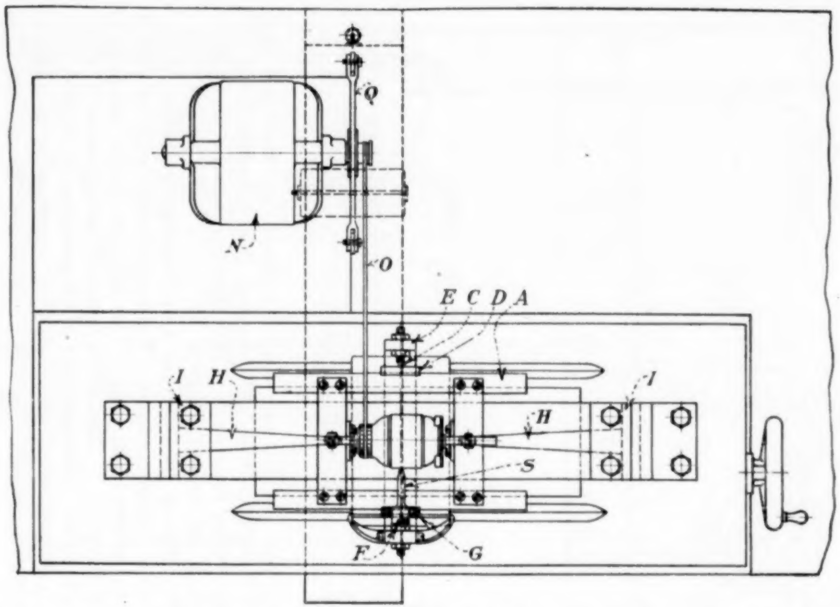


FIG. 5 HORIZONTAL VIEW OF BALANCING MACHINE WITH MOVABLE FULCRUM

moved so that the pointer S is in one of the transverse planes in which the unbalance is to be corrected. The treadle R is pressed down, thus closing the switch P and releasing the brake Q. The rotation of the armature produces vibrations of the bed around the fulcrum axis; the amplitude is a measure of the weight to be applied at the other end of the armature. This weight is recorded by the vibrating reed. The treadle is then released, stopping the driving motor almost instantaneously. The recorded weight is applied at an arbitrary location and the armature is again rotated. The second amplitude will indicate if this arbitrary location is correct or not. Three to four trials usually locate the weight correctly. The carriage is then moved into the plane where the first weight was applied and the same procedure is repeated.

32 Fig. 6 shows the same

machine slightly modified. The vibrating reed is attached to one end of the vibrating bed. As a result of this, the reed does not record the angular deflection of the bed in a uniform measure for all fulcrum locations. This variation was found to be

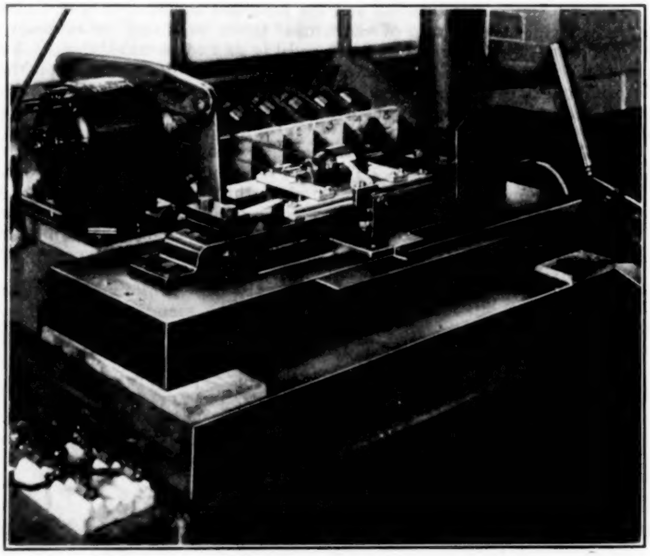


FIG. 6 BALANCING MACHINE, SLIGHTLY MODIFIED

negligible, however. The production of this machine is about twenty armatures per hour at the present time. It may be increased considerably as the operator gains in experience. The accuracy of the balancing result is ± 0.00008 lb. at $\frac{3}{4}$ in. radius, the two balancing planes being 1 in. apart.

APPENDIX NO. 1

CORRECTION OF THE GENERAL STATE OF UNBALANCE BY TWO MASSES

33 When the axis of rotation and the principal axis of inertia of a body do not coincide, the system of centrifugal forces produced by rotation is not in a state of equilibrium and the body is said to be unbalanced. The process of balancing is a change in the distribution of material such that the principal axis is brought into coincidence with the axis of rotation. The displacement of the principal axis may be conveniently segregated into two separate motions:

1 Displacing the principal axis parallel with itself until it intersects the axis of rotation. This can evidently be accomplished by addition or removal of material in the transverse plane[1] of the center of gravity. This part corresponds to static balance, although static balancing generally means balancing on parallel ways, in which case the weight is applied in any transverse plane.

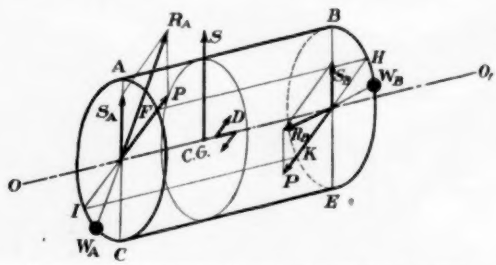


FIG. 7 PERSPECTIVE VIEW OF ROTATING BODY

2 Rotation of the principal axis until it is parallel with the axis of rotation. This corresponds to dynamic balance. It is accomplished by addition or removal of weights on opposite sides of the axis of rotation in the proper longitudinal plane.

34 In other words, the process of balancing brings the center of gravity on the axis of rotation and changes the product of inertia with regard to the proper plane in such a manner that the principal axis becomes parallel to the axis of rotation. When this is obtained the initial system of centrifugal forces has received an addition of a single force and a couple. These counteract the resulting force and the resulting couple of the initial system.

35 It is convenient, therefore, to signify the unbalance by these forces. The force is proportional to the static moment S and the couple is proportional to the product of inertia D. S is the static unbalance, measured in mass times length; D is the dynamic unbalance, measured in mass times length squared. Having these units, the corresponding force and couple are obtained by multiplying S and D, respectively, with the square of the angular velocity. The magnitudes of the centrifugal forces are not, however, of particular interest, so that the mass is frequently replaced by the weight.

36 Fig. 7 shows a rotating body in perspective. The axis of rotation is O–O_1; the principal axis is brought into coincidence with O–O_1 by

[1] Transverse and longitudinal planes in the rotating body represent respectively planes at right angles to and parallel with the axis of rotation.

parallel displacement in the plane *ABCE* and by rotation in the plane *FHIK*. The corresponding static moment and product of inertia are *S* and *D*. Since the couple *D* can be moved to any position in the plane *FHIK*, it may be represented by two vectors *P* in any two transverse planes *AC* and *BE*. The vector *S* may similarly be replaced by its components S_A and S_B in the transverse planes *AC* and *BE*. The vectors S_A and *P* may be replaced by their resultant R_A and, in the same manner, S_B and *P* by R_B. The vectors R_A and R_B are therefore equivalent to *S* and *D*. It follows that the initial state of unbalance may be corrected in the planes *AC* and *BE* by addition or removal of masses W_A and W_B corresponding to the static moments, R_A and R_B.

APPENDIX NO. 2

REQUIREMENTS FOR CONSTANT PERIOD OF BALANCING MACHINE WITH MOVABLE FULCRUM

37 A schematic arrangement is shown in Fig. 8. The vibrating member is shown as a compact body; in reality it includes the bed, the rotor to be balanced, and the driving mechanism. The center of gravity of this aggregate is located in *c*. The weight is carried by the two spring members

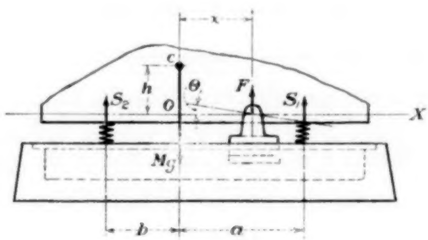


FIG. 8 SCHEMATIC ARRANGEMENT OF BALANCING MACHINE WITH MOVABLE FULCRUM

S_1 and S_2 and the fulcrum member *F*. This consists of a pivoting arrangement that can be moved longitudinally to any desired position. The system is thus capable of performing a vibrating motion around parallel axes in the horizontal plane *O-X*. Strictly speaking, the arrangement has an unlimited number of degrees of freedom; each fulcrum location gives, however, a system with but one degree of freedom.

38 The investigation in this connection will be restricted to the free vibrations of the body. The forced vibrations will be investigated later. It will be assumed, therefore, that the body is displaced from its position of equilibrium by some outside influence, the nature of which it is not necessary to define.

39 The location of the pivot will be referred to a reference plane through the center of gravity at right angles to the direction of the motion of the pivot. The fulcrum axis is always parallel to this plane. Let its distance from this reference plane be x, measured so that x is positive for locations to the right of the reference plane and negative for positions to the left. The spring members are likewise located with reference to this plane, at the distances a and b, respectively. The center of gravity of the vibrating body is located at a distance h above the fulcrum plane *O-X*.

40 For the other quantities, assume the following nomenclature:

M = mass of the vibrating body. (Unit = M)
i_0 = radius of gyration with regard to an axis through the center of gravity, parallel to the fulcrum axis. (Unit = L)
k_1, k_2 = characteristics [1] of the springs. (Unit = MT^{-2})
S_1, S_2 = spring reactions. (Unit = MLT^{-2})
F = reaction on the fulcrum. (Unit = MLT^{-2})
θ = angle of deflection from position of equilibrium in radians. (Unit = 1)
g = acceleration due to gravity. (Unit = LT^{-2})
t = time. (Unit = T)

41 The moment of inertia with regard to the arbitrary fulcrum axis is therefore

$$I = M(i_0^2 + h^2 + x^2) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

42 If the body is displaced through a small angle θ, the restoring moment R_1 is evidently

$$\begin{aligned} R_1 &= S_1(a - x) - S_2(b + x) + Mg(x - h\theta) + \\ &\qquad k_1(a - x)^2\theta + k_2(b + x)^2\theta \\ &= S_1(a - x) - S_2(b + x) + Mgx + \\ &\qquad [k_1(a - x)^2 + k_2(b + x)^2 - Mgh]\,\theta \quad . \; . \; . \; . \; . \; . \quad [2] \end{aligned}$$

43 The first term is independent of θ and represents the restoring moment at rest, hence it must be zero. Therefore

$$\frac{R_1}{\theta} = k_1(a - x)^2 + k_2(b + x)^2 - Mgh = R \quad . \; . \; . \; . \; . \; . \; . \; . \quad [3]$$

This is properly defined as the *restoring factor* for this specific fulcrum location.

44 The system will undoubtedly have a certain amount of friction due to internal friction in the springs, friction in the pivots, air resistance, etc. This will be neglected at the present time, however, but the justification therefor will be evident in the subsequent analysis of the actual motion in Appendix No. 3.

45 D'Alembert's principle may now be applied to the motion. This gives

$$I\frac{d^2\theta}{dt^2} + R\theta = 0 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [4]$$

This represents a harmonic motion, the period of which is

$$T = 2\pi\sqrt{\frac{I}{R}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

Replacing T by the corresponding frequency per minute N, and introducing the values of I and R from [1] and [3],

$$N = \frac{30}{\pi}\sqrt{\frac{k_1(a-x)^2 + k_2(b+x)^2 - Mgh}{M(i_0^2 + h^2 + x^2)}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

46 Assuming k_1, k_2, a, b, M, and h to be constants, N is a function of x. k_1, k_2, a, b, M, and h may, on the other hand, be considered as potentially variable. The problem, then, is to find such a combination of a, b, k_1, k_2, M, and h that will make N independent of x. The mathematical expression for this is

$$\frac{\partial N}{\partial x} = 0 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7]$$

[1] The characteristic of a spring will be understood to mean the force per unit length deflection, that is, the slope of the deflection curve.

47 Equation [7], however, may be replaced by the equivalent equation

$$\frac{\partial(N)^2}{\partial x} = 0 \qquad [8]$$

which is easier to evaluate. Then

$$\frac{\partial(N)^2}{\partial x} =$$

$$\frac{M(i_0^2+h^2+x^2)[-2k_1(a-x)+2k_2(b+x)]-[k_1(a-x)^2+k_2(b+x)^2-Mgh]2Mx}{[M(i_0^2+h^2+x^2)]^2} \qquad [9]$$

48 The denominator is always positive for all values of x. It is sufficient, therefore, to put the numerator equal to zero. This gives, after rearranging,

$$(k_1a - k_2b)x^2 + [(k_1 + k_2)(i_0^2 + h^2) - (k_1a^2 + k_2b^2 - Mgh)]x + (k_1a - k_2b)(i_0^2 + h^2) = 0 \qquad [10]$$

49 Each individual term must be zero if the entire expression is zero for all values of x. This gives three equations, two of which, however, are identical. The two equations determining the required relationship between k_1, k_2, i_0, h, M, a and b, are therefore

$$k_1a = k_2b \qquad [11a]$$

$$(k_1 + k_2)(i_0^2 + h^2) = k_1a^2 + k_2b^2 - Mgh \qquad [11b]$$

These are the necessary and sufficient conditions that make N independent of x.

50 To determine which of k_1, k_2, a, b, M, and h should remain constant in these expressions is a matter of choice and design. For other reasons it is desirable to make h as small as possible. Assume, therefore, that h is kept constant and that k_1, k_2, M, a, and b may be varied at liberty. It is of course evident that this variation should be made in such a manner that the speed is not influenced. There are five variables and only two equations. This means that there are an unlimited number of combinations that satisfy Equations [11a] and [11b]. Putting

$$k_1 = k_2 = k \qquad [12a]$$

and

$$a = b = \rho \qquad [12b]$$

Equations [11a] and [11b] are certain to be satisfied if

$$2k(i_0^2 + h^2) = 2k\rho^2 - Mgh \qquad [13]$$

This gives

$$\rho = \sqrt{i_0^2 + h^2 + \frac{Mgh}{2k}} \qquad [14]$$

51 Equation [6], giving the value of N, makes it possible to check the above result and to obtain the actual value of N. Introducing [12a] and [12b] into [6] gives

$$N = \frac{30}{\pi}\sqrt{\frac{2k(\rho^2 + x^2) - Mgh}{M(i_0^2 + h^2 + x^2)}}$$

With the value of ρ from Equation [14]

$$N = \frac{30}{\pi}\sqrt{\frac{2k}{M}\,\frac{i_0^2 + h^2 + \dfrac{Mgh}{2k} + x^2 - \dfrac{Mgh}{2k}}{i_0^2 + h^2 + x^2}}$$

$$N = \frac{30}{\pi}\sqrt{\frac{2k}{M}} \qquad [15]$$

52 Thus, if the spring members are identical and located symmetrically with reference to the center of gravity of the vibrating system, the natural frequency is independent of the location of the fulcrum and determined by Equation [**15**] if one-half of the distance between the springs is determined by Equation [**14**].

53 A few remarks on the mechanical arrangement are necessary. It is possible to arrange the springs in two different ways.

(*a*) *Springs Under Tension.* In this case the fulcrum member supports the spring reactions in addition to the weight. Apart from the fact that the load on the fulcrum will be great, it means considerable variation in the position of rest for varying fulcrum locations. If the fulcrum should be moved into coincidence with one of the spring members, no equilibrium is possible. The springs have to be spaced quite far apart and the radius of gyration of the bed has to be increased accordingly by adding weight to the bed. Therefore this arrangement is practical only when the fulcrum is to be moved short distances from the central position.

(*b*) *Springs Under Compression.* The fulcrum supports only the balance of the weight not carried by the spring. This may be reduced to zero by proper adjustment of the springs. This arrangement also permits the fulcrum to be moved past the springs if the mechanical arrangement does not obstruct the motion in other respects. The fact that the load on the pivots may be reduced to zero is of especial value because it gives the same position of rest for all locations of the fulcrum. This arrangement has been applied in our development.

54 If the angular velocity corresponding to N be represented by ω,

$$\omega = \sqrt{\frac{2k}{M}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{16}]$$

Introducing this into Equation [**14**] gives

$$\rho = \sqrt{i_0^2 + h^2 + \frac{gh}{\omega^2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{17}]$$

As long as h is small, the first term will be dominating. If, however, h should be made large, the second term will be considerable, as will the third, especially at low speeds.

55 If h is made equal to zero,

$$\rho = i_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{18}]$$

that is, one-half the distance between the springs should equal the radius of gyration of the vibrating systems with reference to the central axis. The same condition occurs if h is negative and of the value

$$h = -\frac{g}{\omega^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{19}]$$

56 When $h = 0$ it is possible to give a simple illustration of the meaning of the principle. If the vibrating system be replaced by two equivalent masses, the arrangement shown in Fig. 2 will be obtained. If the masses are connected by a weightless member, small oscillations about any fulcrum axis represent vertical movements of the two masses. The natural frequency of a mass $M/2$ supported by a spring with characteristic k is

$$N = \frac{30}{\pi}\sqrt{\frac{k}{\frac{M}{2}}} = \frac{30}{\pi}\sqrt{\frac{2k}{M}}$$

which agrees with the result obtained from Equation [**6**].

APPENDIX NO. 3

ANALYSIS OF THE VIBRATING MOTION

57 An investigation of the actual motion of the vibrating system in the balancing machine under the influence of an unbalanced body is of interest, because it enables us to obtain a relation between the unbalance and the amplitude of the motion. It also gives means for predetermining the sensitiveness.

58 Fig. 1 represents the author's arrangement with a movable fulcrum, adapted for constant period. The unbalanced centrifugal forces will produce forced vibrations of the system. The only factor restricting the motion at resonance is the damping. This makes it necessary to investigate more closely the nature of this damping.

59 It appears that the air resistance and the friction at the pivot do not have an appreciable influence upon the motion. The damping effect must therefore be associated with the springs. The exact nature of this property of springs has never been properly investigated. It is certain, however, that the effect is a resistance against *changes in motion*, that is, it is a function of the velocity. For small velocities one is justified in assuming this function as linear. Assume, therefore, that the spring members are such that the resistance against changes in the rate of compression or extension s can be written

$$f = c\frac{ds}{dt} \quad \ldots\ldots\ldots\ldots [\mathbf{20}]$$

when $\frac{ds}{dt}$ is the vertical velocity of the vibrating bed at the spring support. This varies with x, the coördinate of the fulcrum axis, so that the damping moment will also be a function of x. It is found that the total damping resistance from the springs, expressed as a moment with respect to the fulcrum axis, is

$$R_f = c(\rho - x)^2\frac{d\theta}{dt} + c(\rho + x)^2\frac{d\theta}{dt}$$

$$= 2c(\rho^2 + x^2)\frac{d\theta}{dt} \quad \ldots\ldots\ldots\ldots [\mathbf{21}]$$

The damping factor of the motion is therefore

$$C = 2c(\rho^2 + x^2) \quad \ldots\ldots\ldots\ldots [\mathbf{22}]$$

60 The effect of the unbalance may be conceived as a moment on the vibrating mass with respect to the fulcrum axis. The value of this moment is

$$P\omega^2 \sin \omega t$$

where P = unbalance in mass times length squared, with respect to the fulcrum axis. No distinction is made between static and dynamic unbalance. P is the resultant of both and varies with x

ω = angular velocity of rotation.

61 Using the same notation as in Appendix No. 2, the following differential equation may be obtained:

$$I\frac{d^2\theta}{dt^2} + C\frac{d\theta}{dt} + R\theta = P\omega^2 \sin \omega t \quad \ldots\ldots\ldots\ldots [\mathbf{23}]$$

62 The solution of this equation contains two terms, the complementary function and the particular integral. The former will vanish after a certain time interval and only the particular integral will remain. This particular integral has the value

$$\theta = \frac{P\omega^2 \sin(\omega t - \phi)}{\sqrt{C^2\omega^2 + (R - I\omega^2)^2}} \quad \text{[24]}$$

where the phase difference ϕ is determined by

$$\tan \phi = \frac{C\omega}{R - I\omega^2} \quad \text{[25]}$$

If the speed be adjusted so that

$$\omega = \omega_1 = \sqrt{\frac{R}{I}} \quad \text{[26]}$$

that is, so that the frequency of the impressed force equals the natural frequency of the vibrating system *without friction*,[1] then

$$\tan \phi = \tan \phi_1 = \infty$$

that is,

$$\phi_1 = \frac{\pi}{2}$$

hence

$$\theta = \frac{P\omega_1}{C} \sin\left(\omega_1 t - \frac{\pi}{2}\right) \quad \text{(27)}$$

63 The balancing machine under consideration is provided with a device which indicates the motion of a point on the vibrating table. Assume that this indicating device is arranged in such a manner that *the angular deflection* is recorded. If θ_{max} is this amplitude and ω_1 the resonance speed,

$$\theta_{max} = \frac{P\omega_1}{C} \quad \text{[28]}$$

where

$$\omega_1 = \sqrt{\frac{R}{I}}$$

64 Now, if the construction of the vibrating bed is in accordance with the requirements for constant period,

$$\omega_1 = \sqrt{\frac{2k}{M}} \quad \text{[29]}$$

Introducing the value of C from Equation [**22**], there is obtained, for the amplitude,

$$\theta_{max} = \frac{P\omega_1}{2c(\rho^2 + x^2)} \quad \text{[30]}$$

65 The angular deflection of the vibrating table at resonance is directly proportional to the unbalance and varies with the fulcrum location. This variation, however, is symmetrical with reference to the center of gravity of the vibrating system. It is possible, therefore, to arrange the balancing machine in such a manner that the readings for two fulcrum locations are comparable.

[1] This is usually defined as resonance. The kinetic energy is a maximum. The maximum amplitude occurs at a slightly higher frequency of the impressed force. The difference in amplitude is infinitesimal, however.

APPENDIX NO. 4

SENSITIVENESS OF A BALANCING MACHINE

66 This subject has not as yet received a great deal of study, in spite of its importance. It has been shown in Appendix No. 3 that the maximum amplitude is directly proportional to the unbalance P and the angular speed ω_1, and inversely proportional to the damping factor C. From this reasoning it is to be inferred that if the damping could be made sufficiently small, the effect of a certain unbalance would be independent of the vibrating inertia. There is, however, another factor that enters into the sensitiveness of a balancing machine, namely, the time required for the building up of the resonance amplitude. This time element is of no less importance than the final magnitude of the deflection. If it becomes too great, the balancing operation is made very difficult and slow. It is difficult to hold the speed perfectly constant for any considerable length of time, and even small speed variations will prevent the vibrating mass from absorbing energy at the proper rate. It has been found that if the vibrating system is too slow to build up its motion it is impossible to obtain consistent readings of the balancing machine. It is of interest to investigate the factors limiting this.

67 Assume as before that the effect of the unbalance is represented by P, that the moment of inertia of the vibrating system is I, and that the resonance speed is ω_1. The damping factor is C and the amplitude will be indicated by θ.

68 Since the amplitude of the free vibrations of the system diminishes with the expression $e^{-\frac{C}{2I}t}$, the amplitude will grow from zero to its maximum value according to the expression

$$\theta = \theta_{max}(1 - e^{-\frac{C}{2I}t}) \qquad [31]$$

where θ_{max} has the value

$$\theta_{max} = \frac{P\omega_1}{C} \qquad [32]$$

To materialize these conditions, the assumption may be made that the rotor is rotated at the resonance speed while the bed is released for motion when $t = 0$. After t seconds the amplitude will have obtained the value given by Equation [31] [1]. The rate of increase in θ is

$$\frac{d\theta}{dt} = \theta_{max}\,\frac{C}{2I}\,e^{-\frac{C}{2I}t}$$

$$= \frac{P}{I}\cdot\frac{\omega_1}{2}\,e^{-\frac{C}{2I}t} \qquad [33]$$

69 The value of $\frac{d\theta}{dt}$ at the start, that is, when $t = 0$, is evidently a factor of great importance. This is

$$\left(\frac{d\theta}{dt}\right)_{t=0} = \frac{P}{I}\times\frac{\omega_1}{2} \qquad [34]$$

[1] A similar expression will be obtained from the complete solution of the differential Equation [23] in Appendix No. 3 if the integration constants are evaluated with these assumptions. When the reduction of the natural frequency due to damping is neglected the result will be identical with [31].

This is independent of C and may therefore be predetermined without difficulty.

70 The rate of increase of θ at the start is therefore directly proportional to the speed and to the factor P/I. Since the resonance speed varies only slightly for varying weights of rotors, it is evident that the minimum value of the expression P/I is a logical measure of the sensitiveness of a balancing machine. The minimum value of P/I (or the minimum value of P) should be given for each balancing machine because it enables the operator to decide whether or not the result is consistent with the desired accuracy.

DISCUSSION

G. M. Eaton. The use of balancing machines in connection with electrical rotating apparatus has been confined largely to electrical manufacturers. The reasons for this, from the standpoint of the users of electrical machines, included the practically prohibitive first cost of the balancing machine and the high degree of technical skill required for operating it. This skill could hardly be retained, nor could the price be justified in a maintenance shop where the volume of work demanding balance was small. It follows, therefore, that if the balance of a rotor which was dynamically balanced by the builder is disturbed, the user must trust to luck, try to balance in a lathe, have a man from the builder's do the hand balancing, or send the rotor to the builder.

Experience proves that hand balancing is, at best, an approximation, and is an art that few men acquire, while the objections to the first and last methods are obvious. This places an inescapable responsibility on the builder of electrical rotating machinery. A superficial solution would be the avoidance of installations demanding refined dynamic balance for successful operation. But this is no solution, as it would deprive the trade of machines whose functions fundamentally demand the speeds, mountings, etc., which can be properly operated only when dynamic balance is close to theoretical perfection.

The Westinghouse Electric and Manufacturing Company has clearly realized its responsibility, and has been conducting an intensive study of the whole subject. The aim of this study, in which Mr. Söderberg has played an important part, has been, first, to reduce the balancing operation to the simplest procedure in keeping with the fundamental requirements.

The second ideal in this development is to make balancing machines as available as possible to the trade, and, at the same time, to broaden the general knowledge of the far-reaching effects of the vibrations produced by unbalance in rotating machines.

With a growth of the realization of what vibration produces in general maintenance charges, and with the coincident simplification of balancing machines and the art of handling them, balancing machines will be more widely installed. Large operators who now serve many units in a single maintenance shop will find it economi-

cal to use balancing machines in the comparatively near future. In other settings, installations in centrally located service stations will be more logical. An attractive third possibility is offered by portable balancing machines, which to the limit of their feasibility permit the user of rotating machinery to secure balance with a minimum of out-of-service time for his machines, and without the permanent installation of apparatus not in continual use.

In analyzing the services where maintenance may be subject to material reduction by keeping the rotors in correct dynamic balance, the possibility of an interesting general misconception has been brought out. The rougher the fundamental service conditions, the less convincing has been deemed the logic of dynamic balance. It is generally accepted that a more or less definite amount of maintenance must be encountered, and that running balance would be an expensive luxury. Therefore it has been thought best to focus attention exclusively on making the apparatus as rugged as possible, and to accept the resulting maintenance as representing the irreducible overall economy.

A little thought, however, shows that the peak forces of an internal sustained wave disturbance synchronous with the revolutions must frequently fall in phase with maximum disturbing forces of external origin. This combination produces resultant peaks of needless severity. Furthermore the present active researches into fatigue phenomena focus attention sharply on all sustained force cycles. The old beliefs are therefore being shaken, and serious doubt is arising in many minds as to the economy of running dynamically unbalanced rotors in rough-and-tumble services.

Carl A. Johnson. A machine invented to utilize a patented mathematical process invented by Dr. B. L. Newkirk of Schenectady and built by the Gisholt Machine Company is shown in Fig. 9. This machine applies these principles of dynamic and static balancing so that it is possible to obtain complete dynamic and static balancing by two single corrections, individually measured and located near the ends of the body.

A spring-mounted and pivoted frame carries a special type of headstock and adjustable rollers to support the work. The form and location of the springs permit a free vertical vibration of the frame about the pivot springs as a fulcrum. All revolving parts, including the rollers which support the work, are mounted on ball bearings. A speed of rotation above the critical speed of the frame is first used, the driving power then being disengaged, permitting a gradual diminution of speed down to and through the free critical speed of the frame. The dial indicator, mounted upon the column at the left, Fig. 10, indicates the amount of the required correction.

A disk at the left of the revolving parts carries a standardized 10-oz. weight, adjustable radially by means of a vernier reading to 0.01 in. These parts are exactly balanced when the weight is

at zero. The disk is adjustable to any angle by reference to the protractor dial.

With the correction thus arbitrarily applied the machine is again speeded up and allowed to pass through the critical speed. The second amplitude in this process bears a relation to the first amplitude dependent on the angle between the point of application and the point required. After determining and setting off this angle, a third run will check the result. A special calculating rule at the top of the column at the left (Fig. 10), requiring but two settings, determines the corrections proportionate to amplitude and the angles.

After the first determination the work is reversed and the

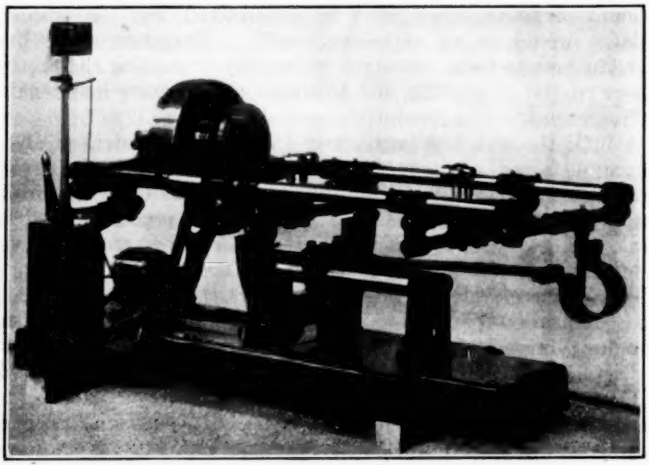


Fig. 9 Precision Balancing Machine

correction determined for the plane initially over the pivots, thus completing the operation. A critical speed of about 100 to 110 r.p.m. is generally employed, which permits visual observation of the amplitudes and prevents distortion of the work from centrifugal forces.

The machine is adapted to a wide variety of work, including balancing crankshafts, flywheels, rotors, pulleys, and other revolving parts. It will receive bodies up to 25 in. in swing and 32 in. between bearings, and within its range is universal in its application. Other sizes outside of this standard form will be built as required.

The physical law utilized in the exact measurement of unbalance by this process is of wonderful consistency and accuracy, so much so that the results of this machine's readings can be checked upon a separate set of scales. In this way it directly establishes facts about which there can be no question and fulfills a long-felt

requirement for a thoroughly reliable and simple instrument or appliance for the precise measurement of rotative unbalance.

Its use affords a complete analysis of the problem of vibration due to rotative unbalance, both as applied to laboratory or to ordinary manufacturing and production propositions, and is one

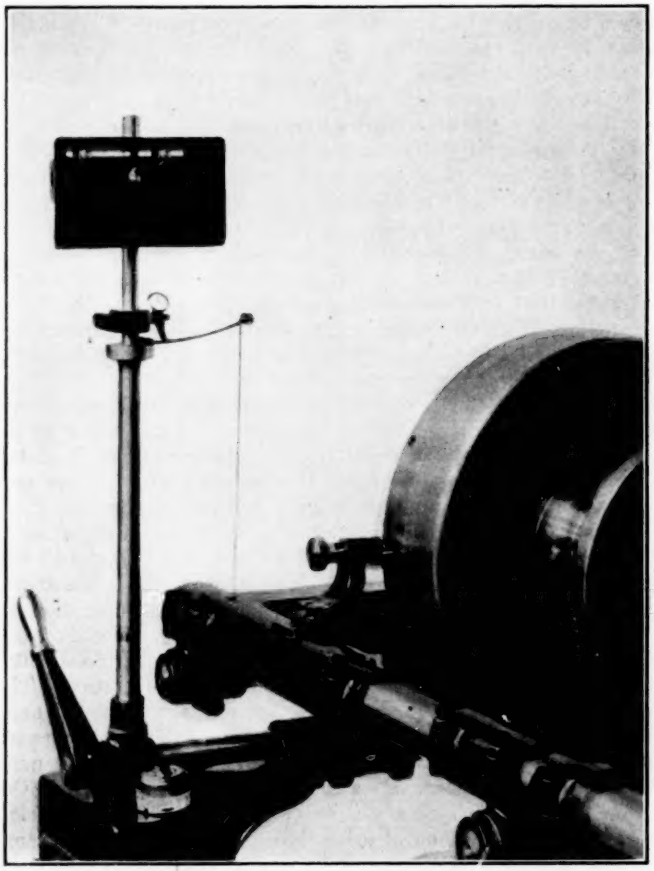


Fig. 10 Dial Indicator and Computation Rule of Gisholt Machine

of the most important of recent contributions to the progress of the mechanical arts.

C. C. Brinton.[1] The author has described a machine that has solved a difficult shop problem, and is a real commercial success. This, however, is only the beginning of development in this field. The problems to be met include rotors up to about 300,000 lb.

[1] Westinghouse Elec. & Mfg. Co., East Pittsburg, Pa.

Old methods of balancing cannot meet the requirements of modern production on account of the high cost and inaccurate results. With the older types of balancing machines, where accurate static balance is necessary before a rotor can be run in the balancing machine, the small, unavoidable errors of the static balance affect the dynamic balance to such a degree that accurate results cannot be obtained. This difficulty may be partly overcome by taking a second run with the rotor turned around. If the two sets of readings do not check, then the proper corrective weight must be calculated from the relation of the two sets of readings. The older machines, however, are handicapped to a certain extent by badly proportioned parts, and much investigation must yet be made to determine these important relations.

A development of the future should be a line of portable balancing machines for use where large production is not a requirement but accuracy is essential. The machines could be made at a reasonable cost if need of a manufacturing quantity can be shown.

A rotor that has been troublesome to balance is of the three-bearing type with two rotors on the same shaft. Our present machines are designed to use only two bearings, and a problem to be solved is the design of one to balance such rotors.

The Author. The Newkirk machine was developed shortly before the machine described in the paper, and the principles are somewhat the same, although the author believes that the latter machine goes one step farther. Dr. Newkirk also arrives at a balancing result without static balance, but it is obtained in somewhat the same way as in the pedestal type of balancing machines. He uses a fixed fulcrum location, and the author believes that he changes the position of the rotor with reference to this. He applies the fulcrum in the balancing plane only in the first operation.

Now, according to the principle mentioned in Appendix No. 1 the remaining unbalance will be a static unbalance in some other plane, which is remedied by a second balancing operation. This second operation is not performed around the first balancing plane, however, but around some other plane. This arrangement retains a few of the disadvantages of the old type of balancing machines. It is, however, a considerable improvement. One reason why the Newkirk method was not applied in our development is that his method of drive does not work for small rotors.

There is a great distinction between the balancing problems of large rotors and of small ones. In the latter case the difficulties are multiplied many times. The machine described in the author's paper is specifically made for very small rotors.

In answer to the question whether a rotor of a centrifugal pump will become unbalanced when working in the fluid, having previously been balanced empty, it is the author's opinion that the difficulty is not one of mechanical balance, but a question of design to be corrected by changing the conditions of flow or some other rearrangement of the design.

No. 1893

BENDING STRESSES IN CURVED TUBES OF RECTANGULAR CROSS-SECTION[1]

By S. Timoshenko,[2] East Pittsburgh, Pa.

Non-Member

This paper analyzes the stresses in bent tubes of rectangular cross-section and shows that in the case of thin tubes the distortion due to bending results in greater flexibility and less strength than given by the usual formulas. In an appendix an approximate solution of the problem is obtained by a consideration of the potential energy of deformation.

The bending of curved tubes is accompanied by a distortion of the cross-section. As a result of this distortion thin tubes are more flexible and have less strength than given by the usual formulas. In one example of a Fairbairn crane, considered later, the maximum stress is 67 per cent greater than the value given by the ordinary formula for the bending of curved bars.

2 This paper will consider the case of the bending of a tube under the action of moments M only. Referring to Fig. 1, if the dimension a of the cross-section is small in comparison with the radius of curvature R of the tube, the maximum stress and the increase of the angle α are usually calculated by the formulas

$$p_{max} = \frac{aM}{2I} \quad \text{. . . . . [1]}$$

$$\Delta\alpha = \alpha\,\frac{MR}{EI} \quad \text{. . . . . [2]}$$

where E denotes the modulus of elasticity and I the moment of inertia of the cross-section about the neutral axis.

3 In the case of a solid cross-section these formulas are sufficiently accurate, but in the case of a tube the problem is more complex. The forces of tension acting on any element ss_1 (Fig. 1)

[1] The case of tubes of circular cross-section has been considered by Prof. Th. Kármán. See *Zeit. Ver. Deutsch. Ing.*, 1911, p. 1889.

[2] Research Engineer, Westinghouse Electric & Manufacturing Co.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers.

and the forces of compression acting on any element rr_1 give resultants whose direction is toward the neutral axis. These forces produce the distortion of the cross-section shown in Fig. 1(*b*).

4 Assuming that the cross-sections of the tube remain plane on bending, the conclusion follows that the elongation e of any element ss_1 will depend not only on the increase $\Delta\alpha$ of the angle α, but also on the radial displacement w due to the distortion of the cross-section.

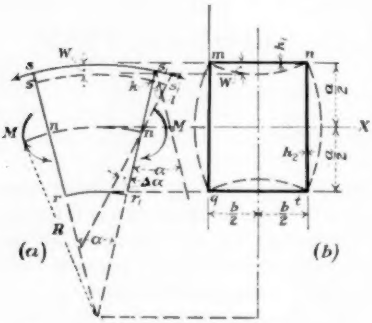


Fig. 1 Section of Bent Tube of Rectangular Cross-Section

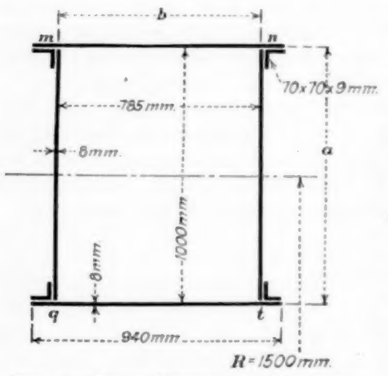


Fig. 2 Cross-Section of Fairbairn Crane Box Girder

5 Let $s's_1'$ represent the position of the element ss_1 after deformation [Fig. 1(*a*)]. It is seen that the extension ls_1' of the element can be represented by

$$ls_1' = ks_1' - kl$$

6 The first term on the right-hand side of this equation, due to rotation of the cross-section s_1r_1 about the neutral axis X, is equal to $\frac{\Delta\alpha a}{2}$; w is assumed to be small in comparison with a. The second term, due to radial displacement w, is equal to $w\alpha$. Substituting this value in the equation and dividing it by $R\alpha$, the initial length of the element ss_1 (a assumed to be small in comparison with R), the following expression is obtained for the longitudinal strain of the element ss_1:

$$e = \frac{\Delta\alpha a}{2R\alpha} - \frac{w}{R} \quad . \quad . \quad . \quad [3]$$

Formula [**3**] may also be used in calculating the compression of any element rr_1. It is seen that as a result of the distortion of the cross-section, the stresses at the middle of the plates *mn* and *qt* become less than those given by Formula [**1**]. This decrease of stresses in the central portions of *mn* and *qt* will necessarily be associated with an increase of stresses in other parts of the cross-section.

7 The true values of the maximum stress and of the increase

of the angle α may be obtained by using Formulas [**1**] and [**2**] if for I there be substituted a smaller quantity

$$I_1 = \beta I \qquad [4]$$

where the coefficient β, less than unity, is to be calculated by the following formula (see Appendix):

$$\beta = 1 - 2\Lambda \times \frac{12(1-\sigma^2)\left[2 + \frac{1}{2}\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3\right]^2}{\pi^2\left(1+\frac{i_2}{i_1}\right)\left[\pi^4 + 4\pi^2\frac{b^2}{a^2}\left(\frac{h_2}{h_1}\right)^6 + \frac{40}{3}\pi^2\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3\right] + 12\Lambda(1-\sigma^2)\pi^2\left(1+\frac{i_2}{i_1}\right)\left[1 + \frac{3}{4\pi^2}\frac{b^2}{a^2}\left(\frac{h_2}{h_1}\right)^6 + \frac{16}{3\pi^2}\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3\right]} \qquad [5]$$

where h_1 = thickness of horizontal plates mn and qt

h_2 = thickness of vertical plates mq and nt

i_1 = moment of inertia about neutral axis of the horizontal plates

i_2 = moment of inertia about neutral axis of the vertical plates

σ = Poisson's ratio of the material (in these calculations $\sigma = 0.3$)

and

$$\Lambda = \frac{b^4}{R^2 h_1^2} \qquad [6]$$

8 In the case of a tube of square cross-section and of a constant thickness h,

$$a = b;\ h_1 = h_2 = h;\ \frac{i_2}{i_1} = \frac{ha^3}{6} : \frac{ha^3}{2} = \frac{1}{3}$$

and from Formula [**5**]

$$\beta = \frac{49.18 + 1.332\Lambda}{49.18 + 3.232\Lambda} \qquad [7]$$

9 The effect of distortion of cross-section depends on the magnitude of the quantity Λ. If Λ is small, i.e., in the case where the radius R and the thickness h are large, the coefficient β will be nearly unity and Formulas [**1**] and [**2**] will give sufficiently accurate results. Taking another extreme case and putting $\Lambda = \infty$ in [**7**], $\beta = 0.412$. In this case the maximum stresses and the flexibility of the tube are about 2.5 times greater than the values given by Formulas [**1**] and [**2**].

10 As an example, take $\frac{R}{a} = 10$; $\frac{a}{h} = 50$. Then from [**6**] $\Lambda = 25$, which, substituted in [**7**], gives $\beta = 0.513$. The maximum stresses in this case will be about twice as great as given by Formulas [**1**] and [**2**].

11 As a second example consider the cross-section [1] represented in Fig. 2. In this case

$$\frac{h_1}{h_2} = 1; \ \frac{b}{a} = 0.785; \ \Lambda = 2640$$

The connecting angles and the external parts of the horizontal plates have no appreciable influence on the distortion of the cross-section, therefore the moment of inertia of their cross-section is included in the moment of inertia i_2. In this way

$$i_1 = 319 \times 10^3\text{cm.}^4; \ i_2 = 307 \times 10^3\text{cm.}^4; \ \frac{i_2}{i_1} = 0.96$$

Substituting this in [**5**] gives $\beta = 0.60$. It is seen that in this case the maximum stresses and the flexibility of the tube are $1 : 0.60 = 1.67$ times those given by Formulas [**1**] and [**2**].

APPENDIX

12 The approximate solution of the problem of the bending of a curved tube may be obtained from a consideration of the potential energy of deformation. For a given increase $\Delta\alpha$ of the angle α (Fig. 1) the form of distortion of the cross-section of the tube must be such as to make the potential energy of deformation a minimum. The deflection of the plates mn and qt [Fig. 1(b)] may be stated in the form

$$w = w_0 \sin\frac{\pi x}{b} + w_1\left(1 - \cos\frac{2\pi x}{b}\right) \quad . \ . \ . \ . \ . \ . \ . \ . \quad [\mathbf{8}]$$

where w_0 and w_1 are constants to be determined. One relation between these constants may be obtained from a consideration of the deformation of an element of the tube enclosed between two cross-sections unit distance apart.[2] Considering this element as a frame with rigid connections at the points m, n, q, and t, and taking in account the fact that the elastic curves for the verticals mq and nt are arcs of circles, the curvature of verticals will be

$$\frac{2}{a}\left(\frac{dw}{dx}\right)_0$$

and the bending moment

$$\frac{Eh_2^3}{(1-\sigma^2)12}\,\frac{2(dw)}{a(dx)_0} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad [\mathbf{8a}]$$

The bending moments at m and n of the horizontal element mn are

$$\frac{Eh_1^3}{(1-\sigma^2)12}\,\frac{(d^2w)}{(dx^2)_0} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad [\mathbf{8b}]$$

The term $E/(1-\sigma^2)$ is taken instead of E due to the fact that the distortion of cross-sections at bending of $mnpq$ is prevented by the neighboring elements. Equalizing [**8a**] and [**8b**],

$$\left(\frac{dw}{dx}\right)_{x=0} = \frac{a}{2}\left(\frac{h_1}{h_2}\right)^3\left(\frac{d^2w}{dx^2}\right)_{x=0}.$$

---

[1] From description of a Fairbairn crane in Ad. Ernst's Die Hebezeuge, vol. iv, p. 540.

[2] The small angle between the cross-sections is neglected.

Substituting for w the value given in [8],

$$w_1 = w_0 \frac{b}{2\pi a}\left(\frac{h_2}{h_1}\right)^3$$

Then [8] becomes

$$w = w_0\left[\sin\frac{\pi x}{b} + \frac{b}{2\pi a}\left(\frac{h_2}{h_1}\right)^3\left(1 - \cos\frac{2\pi x}{b}\right)\right] \quad . \; . \; . \; . \; . \quad [9]$$

13 The potential energy of deformation of the element *mnqt* consists (*a*) of the potential energy of extension and compression in the direction perpendicular to the plane *xy*, and (*b*) of the potential energy of bending of the frame *mnqt*. The extension of the plate *mn* and the compression of the plate *qt* are represented by Formula [3]. The deformation in the vertical plates *mq* and *nt* follows the linear law:

$$e = \frac{\Delta\alpha y}{R\alpha}.$$

14 The part of the potential energy due to extension and compression will be

$$A_1 = Eh_1\int_0^b\left(\frac{a}{2}\frac{\Delta\alpha}{R\alpha} - \frac{w}{R}\right)^2 dx + Eh_2\int_{-\frac{a}{2}}^{+\frac{a}{2}}\left(\frac{y\Delta\alpha}{R\alpha}\right)^2 dy$$

Substituting [9] for w,

$$A_1 = Eh_1\left\{\left(\frac{a}{2}\frac{\Delta\alpha}{R\alpha}\right)^2 b - \frac{a\Delta\alpha}{R^2\alpha}w_0\left[\frac{2b}{\pi} + \frac{b^2}{2\pi a}\left(\frac{h_2}{h_1}\right)^3\right] + \right.$$

$$\left.\frac{w_0^2}{R^2}\left[\frac{b}{2} + \frac{3}{8}\frac{b^3}{a^2\pi^2}\left(\frac{h_2}{h_1}\right)^6 + \frac{8}{3}\frac{b^2}{a\pi^2}\left(\frac{h_2}{h_1}\right)^3\right]\right\} + Eh_2\left(\frac{\Delta\alpha}{R\alpha}\right)^2\frac{a^3}{12} \quad . \; . \quad [10]$$

15 The potential energy due to bending of the frame *mnqt* will be

$$A_2 = \frac{Eh_1^3}{12(1-\sigma^2)}\int_0^b\left(\frac{d^2w}{dx^2}\right)^2 dx + \frac{Eh_2^3}{12(1-\sigma^2)}\left(\frac{d^2w}{dx^2}\right)^2_{x=0}\left(\frac{h_1}{h_2}\right)^6 a$$

The first term on the right side represents the potential energy due to the bending of elements *mn* and *qt*. The second term corresponds to the bending of *mq* and *nt*.

Substituting [9] for w,

$$A_2 = \frac{Eh_1^3w_0^2}{12(1-\sigma^2)}\left[\frac{\pi^4}{2b^3} + \frac{2\pi^2}{a^2b}\left(\frac{h_2}{h_1}\right)^6 + \frac{8}{3}\frac{\pi^2}{ab^2}\left(\frac{h_2}{h_1}\right)^3\right] + \frac{Eh_2^3}{12(1-\sigma^2)}\,\frac{4\pi^2w_0^2}{ab^2} \quad [11]$$

16 The complete expression for the potential energy of the element *mnqt* will be

$$A = A_1 + A_2$$

17 Calculating the derivative $\frac{dA}{dw_0}$ and putting it equal to zero,

$$w_0\left\{\pi^4 + 4\pi^2\frac{b^2}{a^2}\left(\frac{h^2}{h_1}\right)^6 + \frac{40}{3}\pi^2\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3 + 12(1-\sigma^2)\frac{b^4}{R^2h_1^2}\left[1 + \frac{3}{4\pi^2}\frac{b^2}{a^2}\left(\frac{h_2}{h_1}\right)^6 + \right.\right.$$

$$\left.\left.\frac{16}{3\pi^2}\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3\right]\right\} = 12(1-\sigma^2)\frac{b^4}{R^2h_1^2}\,\frac{a\Delta\alpha}{\alpha}\left[\frac{2}{\pi} + \frac{1}{2\pi}\frac{b}{a}\left(\frac{h_2}{h_1}\right)^3\right] \quad . \; . \quad [12]$$

from which w_0 may be calculated. Substituting this in the equation

$$A = \frac{1}{2}M\frac{\Delta\alpha}{R\alpha} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [13]$$

which is based on the fact that the potential energy is equal to the work done by the external forces,

$$M = EI \frac{\Delta\alpha}{R\alpha}\beta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{14}]$$

where β denotes Formula [**5**] of the paper.

18 In the case of a tube of square cross-section and of constant thickness,

$$h_1 = h_2; \; a = b$$

Equation [**12**] can then be written

$$w_0\left[\pi^4 + \frac{52}{3}\pi^2 + 12(1 - \sigma^2)\left(1 + \frac{73}{12\pi^2}\right)\Lambda\right] = \frac{5 \times 12}{2\pi}(1 - \sigma^2)\frac{a\Delta\alpha}{\alpha}\Lambda$$

from which

$$w_0 = \frac{a\Delta\alpha}{\alpha}\,\frac{0.796\Lambda}{24.6 + 1.62\Lambda}$$

When w_0 is determined, with the aid of [**9**] and [**3**] the elongation e at every point of the plate mn may be calculated. For example, at the middle of the plate mn $\left(x = \frac{b}{2}\right)$,

$$w = w_0\left(1 + \frac{1}{\pi}\right) = 1.318\, w_0$$

$$e = \frac{a}{2}\frac{\Delta\alpha}{R\alpha}\left(1 - \frac{1.592 \times 1.318\Lambda}{24.6 + 1.62\Lambda}\right)$$

It will be seen that as Λ increases the elongation e at this point diminishes, and that if Λ is a large quantity e may become negative.

No. 1894

ENDURANCE-TEST DATA AND THEIR INTERPRETATION

By K. Heindlhofer,[1] Philadelphia, Pa.
Member of the Society

and

H. Sjövall,[1] Philadelphia, Pa.
Non-Member

The determination of the endurance of different products is a subject that hitherto has been neglected, yet it is important. It is a general characteristic of products of nature as well as manufactured products that their life varies within wide limits, and it is thus impossible to judge the life of a certain type of product by a single observation.

The average life determined by repeated endurance tests is a valuable characteristic of comparison as it shows the general magnitude of life, but taken alone it is insufficient to define durability. It must be amended by a second quantity expressing the dispersion or range of variation of the individual lives, which is an indication of reliability. Since the number of available test data is limited, its average is approximate, and its precision will depend on the number of data.

The object of the authors in preparing this paper was to develop methods for computing the probable error of the average life of an object or product depending on the number of repeated tests. This error is a measure of the reliability of the average. The results are presented in the form of a diagram which shows this probable error at a glance. Finally, it is suggested that endurance curves may be employed to determine whether an elimination test will be advantageous in the case of products made in quantity.

THE LIFE of all objects, and especially that of machine parts, varies widely. This is a general characteristic and applies even to objects which apparently are identical, i.e., made in the same factory from the same material, and are used or tested under identical conditions. It is not the authors' intention to investigate the cause of this variability, but rather to represent and to interpret the life or endurance of such objects by figures which may serve as a basis of comparison when the selection or application of machines or machine elements is under consideration.

2 It is customary to express capacity or performance by figures. We speak of the horsepower of a motor, the watt consumption of

[1] Research Engr., S. K. F. Industries.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers.

a lamp, the gallons-per-minute capacity of a pump, these figures indicating whether the machine in question is suitable for the purpose or worth the price; and it would be equally valuable if we could express how long a gear, a bearing, or a die would stand up under given conditions.

3 It is not customary to specify life data on machines. This is partly due to the great difficulty and expense of acquiring such

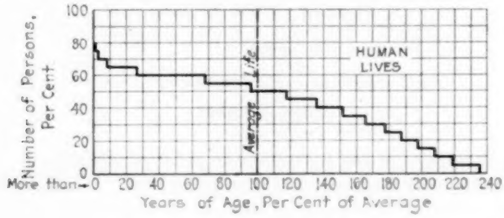


Fig. 1

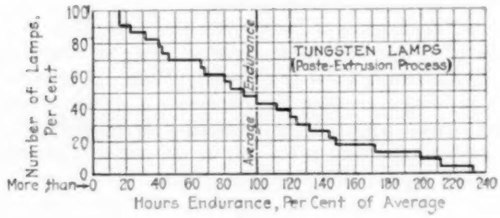


Fig. 2

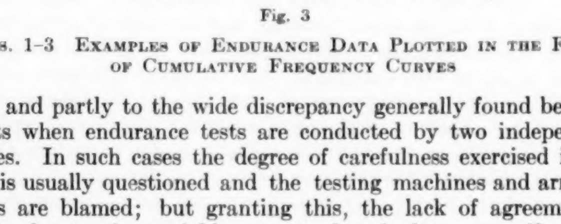


Fig. 3

FIGS. 1–3 EXAMPLES OF ENDURANCE DATA PLOTTED IN THE FORM OF CUMULATIVE FREQUENCY CURVES

data, and partly to the wide discrepancy generally found between results when endurance tests are conducted by two independent parties. In such cases the degree of carefulness exercised in the tests is usually questioned and the testing machines and arrangements are blamed; but granting this, the lack of agreement is mostly due to the variable nature of such data, even if all the conditions are uniform. The discrepancy is due to the inherent variability of endurance in general.

A Method of Interpreting Endurance Data

4 In spite of this difficulty it is possible to derive endurance figures having a comparative value. This may be achieved by collecting a great number of endurance data on products of the same type and size, tested under identical conditions, and arranging the lives in an ascending sequence, the cumulative number of individuals being the ordinates and the lives the corresponding abscissas. Examples taken from actual observations are shown in Figs. 1, 2, and 3. These graphs are called "cumulative frequency curves."

5 It is advantageous to standardize the total length of the

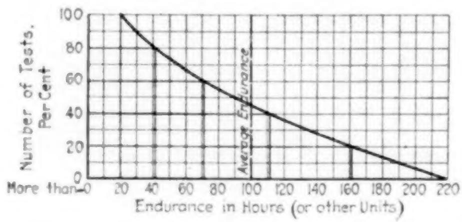


Fig. 4 Representation of Numerous Test Data Obtained Under Identical Test Conditions

ordinate, calling it 100 per cent, and subdivide it as shown in Fig. 4. One hundred per cent would then mean the whole number of individual objects tested. It is easy to pick from the diagram the percentage of all the objects tested which have a life longer (or shorter) than a given number of hours, revolutions, or other units. This idea is illustrated in detail in Fig. 4 and Table 1.

6 In order to be able to obtain these percentages with a suffi-

TABLE 1 REPRESENTATION OF TEST RESULTS PLOTTED IN FIG. 4

Percentage of total number of test objects	Endurance in hours, revolutions, or other units, more than
100	20
90	29
80	41
70	55
60	71
50	90
40	111
30	135
20	161
10	189
0	220

cient degree of accuracy, an extensive collection of data is required. The number in each interval, say, between 10 and 20 per cent must be considerable. Such, however, is seldom the case, for the expense of testing often forbids the collection of more than 10 or 20 data. Such a limited number cannot be split up into intervals

upon the content of which dependence can be placed, therefore it is preferable to choose a characteristic which is a resultant of all the values available. The average or mean of all the values conforms to this requirement, and at the same time it is readily derived and its meaning simple. In many cases knowledge of the average life is sufficient. For example, when a large plant or a city has to be illuminated by incandescent lamps, the cost of replacing the lamps may be determined from the average life of the lamps.

7 However, there are other cases where the average alone, while important, is insufficient to enable one to judge of the merit

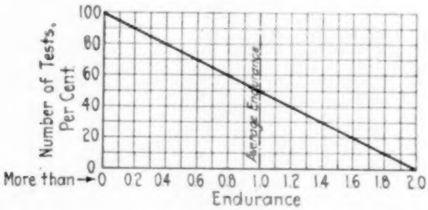


Fig. 5

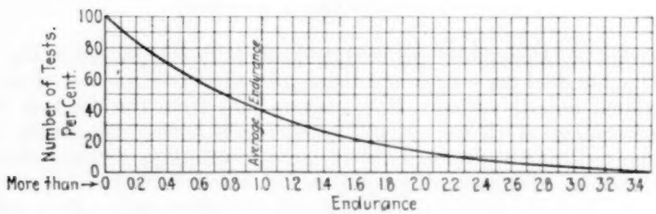


Fig. 6

Figs. 5 and 6 Endurance Figures of the Same Average but of Different Dispersions

of the product. All the elements of construction from which a high degree of reliability is expected must show as small a fluctuation in endurance as possible. For example, the material of a given part of an aeroplane should, in addition to a high average, show as little fluctuation as possible. Thus the deviations from the average must not be too great. This requirement would exclude extremely weak specimens.

8 From this it follows that at least two characteristics, the average life and the variation of lives, are required to define endurance. The difference between the two is best illustrated by the examples presented in Figs. 5 and 6, which show two cumulative curves with identical averages but with different ranges of individual lives. This range or "dispersion" may best be expressed by

the "root-mean-square deviation" of lives from the average, because this value is mainly determined by large deviations. The root-mean-square deviation is obtained by finding the sum of the squares of individual deviations from the mean, dividing this sum by the number of individuals, and extracting the square root.

9 When introducing the average from a small number of tests which vary between wide limits, a further question arises, namely, how reliable is this value? To answer this question, select, say, 10 data at random from an extensive collection, determine their average, and repeat this process many times. When inspecting the large number of averages thus obtained it will be found that their frequency of occurrence will be greater near the mean of the *whole* collection, and that only a few will be found which deviate greatly from the total mean. Two factors govern their distribution. The

TABLE 2 DATA OF CUMULATIVE FREQUENCY CURVE USED IN EXPERIMENT ON VARIABILITY OF AVERAGE OF ENDURANCE VALUES (SEE FIG. 6)

Total average endurance of an infinite number of tests = 100
Root-mean-square deviation from average = 83.9

Endurance values in 50 intervals				
2	26	60	102	174
4	29	64	108	184
6	32	68	114	195
9	35	72	119	207
11	38	76	125	219
14	42	80	132	234
16	45	84	139	251
18	48	89	146	274
21	52	93	154	301
24	56	98	163	347

first is the number of data (in the present example 10) included in a group, and the second the dispersion or range of the individual values in the original collection. It is obvious that a more homogeneous collection (Fig. 5) yields group averages lying close together, while the opposite is the case with less homogeneous data (Fig. 6).

10 For the underlying purpose of the work it would be convenient to set limits within which a given percentage of all possible group averages would be located. Other things being equal, the closer the specified limits, the smaller the percentage of group averages within those limits. This statement may also be expressed in another way: the narrower the limits, the smaller the probability of the appearance of average values within them.

11 The law governing limits when a limited number of data of known dispersion are available, will be illustrated first by an example. The deduction of this law is given later in the paper.

12 Assume that a not too small number of data have been obtained and tabulated (Table 2), and which, when graphically presented, appear as shown in Fig. 6. Their average value is 100. The root-mean-square deviation from the average is

$$\mu = \sqrt{\frac{98^2 + 96^2 + 94^2 + 91^2 + \ldots}{50}} = 84$$

It is now possible to determine the limits $\pm\Delta$ within which the average value of a limited number of tests on the same product will be located with a probability of, say, 50 per cent. The procedure to determine the limits for any other value of the probability, such as 90 per cent or even 99 per cent, would be similar.

13 The logarithmic chart forming Fig. 7 renders the computation simple. First the number of test data is located on the lower bounding line of the chart. Let this number be $n = 10$. The intersection of the ordinate of $n = 10$ with the line $P = 0.50$ (i.e., probability = 50 per cent) gives the value $\pm\dfrac{\Delta}{\mu} = 0.213$. Since the root-mean-square deviation μ is equal to 84, $\pm\Delta = 0.213 \times 84 = 18$.

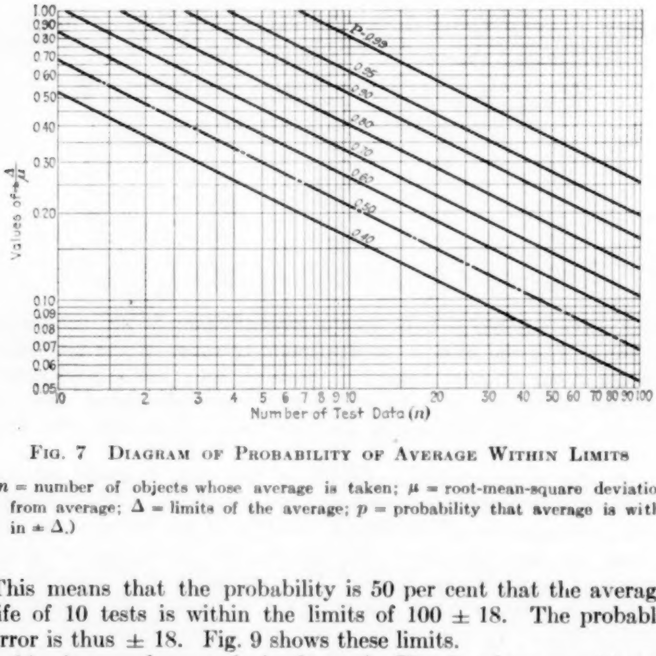


FIG. 7 DIAGRAM OF PROBABILITY OF AVERAGE WITHIN LIMITS

(n = number of objects whose average is taken; μ = root-mean-square deviation from average; Δ = limits of the average; p = probability that average is within $\pm\Delta$.)

This means that the probability is 50 per cent that the average life of 10 tests is within the limits of 100 ± 18. The probable error is thus ± 18. Fig. 9 shows these limits.

14 A second example is shown in Fig. 8, where $n = 10$ and $\mu = 58$. The average is 100 ± 12 at 50 per cent probability. A comparison of Figs. 8 and 9 shows the influence of the dispersion on the limits.

15 The probability may be visualized. Imagine the test series repeated a great number of times. Each series has a different average. In case of the 90 per cent probability, 90 per cent of all the averages would lie inside the limits and 10 per cent outside. In other words, the chances that the average of any of the series

will lie inside the limits are 9 to 1. Similarly, the 50 per cent probability represents equal chances, for which reason the magnitude between the limits is called the probable error.

Verification of the Proposed Method

16 The procedure for determining the probable error of the average, and the more general case of setting limits which include the average at any desired probability, having been described, it

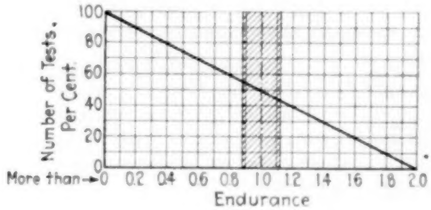


Fig. 8

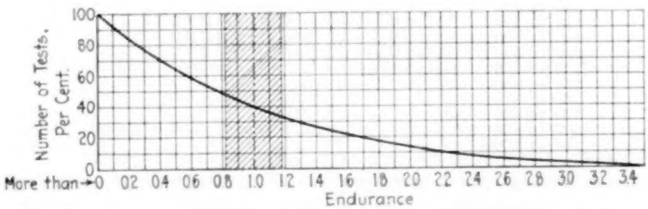


Fig. 9

Fig. 8 and 9 Limits Including 50 Per Cent of All Possible Averages of 10 Values Picked at Random from the Cumulative Frequency Curves of Figs. 5 and 6

remains to be shown how the values used in plotting Fig. 7 were obtained.

17 From the theory of least squares it is known that the probability P of an error $\pm\,\Delta$ of the average value of n observations may be expressed by the formula

$$P_{\pm\Delta} = \frac{2}{\sqrt{\pi}} \int_0^{\frac{\sqrt{n}}{\sqrt{2}}\frac{\Delta}{\mu}} e^{-t^2}dt \quad . \; . \; . \; . \; . \; . \; . \; [1]$$

where μ is the root-mean-square deviation. This formula has been derived under the assumption that the errors of individual observations occur at random, i.e., that they follow Gauss's law of frequency of error.

18 Fig. 7 is based on Formula [1] and is constructed as follows: The number of tests n and Δ/μ are arbitrarily assumed, thus determining the upper limit of the integral. The integral may then be evaluated by tables found in handbooks. The corresponding values of Δ/μ, n, and P may then be plotted, preferably on logarithmic paper.

19 It remains to be shown, however, that Formula [1] is applicable with sufficient accuracy to cumulative frequency curves (Figs. 1, 2, 3, and 4) which do not necessarily follow Gauss's law.

TABLE 3 EXPERIMENT ON THE VARIABILITY OF THE AVERAGE VALUE OF A LIMITED NUMBER OF DRAWINGS

(Each score is the average of 10 drawings from the numbers given in Table 2.)

Percentage of ideal average	Distribution of 250 scores
0 to 10	
10 to 20	
20 to 30	
30 to 40	1
40 to 50	1
50 to 60	11111 11
60 to 70	11111 11111 11111 1
70 to 80	11111 11111 11111 11111 11111 1111
80 to 90	11111 11111 11111 11111 11111 11111 1111
90 to 100	11111 11111 11111 11111 11111 11111 111
100 to 110	11111 11111 11111 11111 11111 11111 11111 11
110 to 120	11111 11111 11111 11111 11111 11111 11
120 to 130	11111 11111 11111 11111 11111 1
130 to 140	11111 11111 11111 1
140 to 150	11111 111
150 to 160	11111
160 to 170	1111
170 to 180	1
180 to 190	
190 to 200	

20 A general proof based on the symbolic method appears to be difficult because the shapes of the cumulative curves are arbitrary. For the purpose in hand it was considered sufficient to apply Formula [1] to two cumulative curves which represent typical conditions such as obtain in practice. In order to broaden the limits of the proof, extreme conditions have been chosen.

21 In the first example a cumulative curve is assumed the asymmetry of which with respect to the average is pronounced (Fig. 6 and Table 2). The endurance values which define this curve were derived from actual test series. In the second example the curve is a straight line (Fig. 5). This symmetrical distribution corresponds to the so-called problem of De Moivre.[1]

22 The proof consists in computing the probabilities by a method the correctness of which cannot be questioned. These values are then to be compared with those derived by using Formula [1], when the comparison will bear out any possible discrepancy. While for the first example an experimental method was used, the second was calculated according to a rigorously correct formula.[1]

[1] E. Czuber, Wahrscheinlichkeitsrechnung, 3d ed., pp. 63–66, Leipsic; B. G. Teubner.

23 The experimental procedure was as follows: The figures given in Table 2 were printed on small wooden disks and collected in an urn. A disk was drawn therefrom and its number recorded. This disk was then returned and the contents of the urn were thoroughly and systematically mixed to insure a random appearance of the disks. The procedure was then repeated a great number of times (2500). Figures thus obtained were arranged in groups of $n = 10$ in the sequence they appeared. Each group was averaged and the results arranged as in Table 3. The frequency of average within a certain interval may be found from this tabulation. The probabilities thus obtained and the corresponding values calculated by using Formula [**1**] are compiled in Table 4.

24 In case of the straight-line distribution (Fig. 5) the probabilities for given limits and 10 tests per group were calculated from De Moivre's formula. These figures are also entered in Table 4 with the corresponding values calculated from Formula [**1**].

25 The agreement between the corresponding figures is satisfactory, justifying the approximation involved in the use of Formula [**1**], which serves to determine the limits of averages such as those derived from endurance-test data. The agreement would be still better for $n>10$, while for $n<10$ the accuracy would necessarily be lower.

Improving the Endurance of Quantity Products by Elimination Tests

26 In addition to their function of gaging quality and reliability, cumulative endurance curves may be applied in a constructive manner. Imagine a quantity product having a cumulative frequency curve like that of Fig. 1. It is evident that this product

TABLE 4 VERIFICATION OF FORMULA [1] FOR 10 DRAWINGS

(*a*) Straight Line, Fig. 5 (Root mean square = 0.577; average = 1.000)

△ = Limit on either side of the average	Probability: Calculated according to Formula [1] or Fig. 7	Probability: Mathematically correct values
± 0.1	0.416	0.412
0.2	0.727	0.722
0.3	0.900	0.900
0.4	0.967	0.973
0.5	0.994	0.995
0.6	0.999	0.999

(*b*) Cumulative Frequency Curve, Fig. 6 (Root mean square = 0.893; avg. = 1.000)

△ = Limit on either side of the average	Probability: Calculated according to Formula [1] or Fig. 7	Probability: Experimentally determined from 250 scores at 10 drawings. (See Table 3.)
± 0.1	0.294	0.280
0.2	0.549	0.544
0.3	0.742	0.764
0.4	0.868	0.892
0.5	0.940	0.952
0.6	0.976	0.976
0.7	0.992	0.996
0.8	0.997	1.000
0.9	0.999	1.000

would be considerably improved by subjecting it to an endurance test resembling service conditions for about one-tenth of the time of the average life before the product left the factory. By this procedure 35 per cent would be eliminated, improving the average of the remainder by approximately 50 per cent.

27 An improvement is not always effected by running all articles through such a test. For example, the products dealt with in Figs. 2 and 3 would lose in value by this process.

28 A judgment as to whether an elimination test will be advantageous or not, can only be based on a careful study of cumulative endurance data.

Conclusions

There are three characteristic figures which define the endurance quality of a product.

(1) The average life, which shows the general magnitude of life

(2) The dispersion or root mean square, which expresses reliability of the *product* itself

(3) The probable error of the average. This indicates the reliability of the *average;* in other words, it shows whether or not the test has been repeated a sufficient number of times.

No. 1895

STEEL-CAR CONSTRUCTION AT THE ANGUS SHOPS OF THE CANADIAN PACIFIC RAILWAY, MONTREAL

By H. R. Naylor,[1] Montreal, Canada

Non-Member

In 1909 the Canadian Pacific Railway, to meet the increasing severity of modern traffic requirements, originated a box car having its entire frame built of steel. The production of such equipment in quantity necessitated the erection of an additional shop for the fabrication of the steelwork, and the structure built embodied in its arrangement many novel features for the rapid handling of material to and from the machines and also during the various stages of assembly.

The present paper describes this shop very completely, giving particulars regarding its layout, crane facilities, machine equipment, etc. The various machining operations and the jig method of car assembly, first put into practice at the Angus shops, are presented in detail, and the methods used in the final erection and finishing of cars are dealt with at some length.

A COMPARISON of the modern 60-ton steel-frame box car with the 30-ton wood-frame box car commonly built fifteen years ago brings out two points: the complete change in design, and the effect this change in design must have had on the car shops. It is obvious that the facilities for building the wood-frame car would be quite inadequate for the steel car and must have created new problems with which the car builder had to contend.

2 The gradual increase in locomotive capacity, making possible the hauling of longer and heavier trains, subjected the wooden cars to severe service conditions, which were in turn met by steel underframes and other steel reinforcements; and at each stage of advancement wood framing was gradually replaced by steel, until in 1909 the Canadian Pacific Railway originated a box car having the entire frame built of steel.

3 The earlier type of steel-frame box car was built with center sills of 15-in. channels, side sills of 8-in. channels, side and end

[1] Asst. Works Mgr., Angus Car Shops, Canadian Pacific Railway.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers. Abridged.

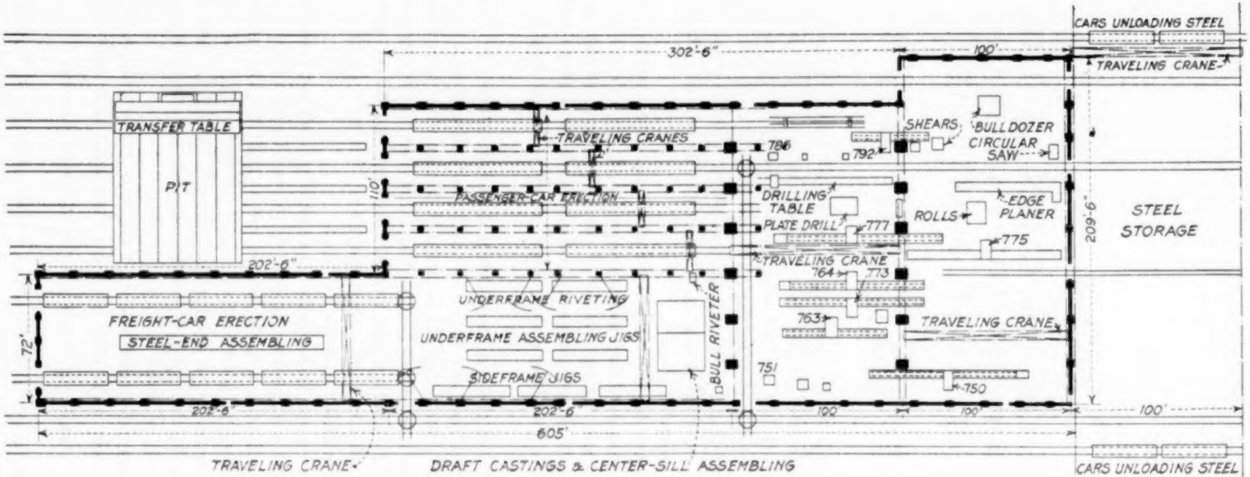


Fig. 1 Plan of Steel-Car Shop, Canadian Pacific Railway Angus Shops, Montreal

posts of 4-in. Z-bars, and corner posts of 5-in. angles. The longitudinal sheathing was bolted to the inside of the framing.

4 The Angus shops were already well equipped for building passenger and freight cars on an extensive scale, but the introduction of the steel-frame box car necessitated the erection of an additional shop. This shop was designed for building the steelwork of both passenger and freight equipment, and embodied in its arrangement many novel features for the rapid handling of material to and from the machines, and also during the various stages of assembly. At the time

of its erection it probably represented the best practice on this continent, the good features of other shops being combined with the original ideas developed at the time the layout was planned.

5 The freight section of the shop was designed to construct steel-frame box cars in the most economical manner, and although some minor modifications have been made in the machinery layout and erecting equipment to meet subsequent developments, the shop has well served the purpose for which it was intended and has furthermore proved equally satisfactory for building large orders of automobile, refrigerator, coal, and flat cars, in addition to steel snow plows and other snow-fighting equipment. At intervals between new car construction the shop is used extensively for repairing and rebuilding steel freight and passenger cars.

Description of the Shop

6 The new shop (Fig.1) faces a midway upon either side of which are also located the supplementary shops, the midway being served with overhead traveling cranes. It is a steel-frame structure, with steel columns carried on concrete piers, the lower foundation walls being of concrete, to a height of $2\frac{3}{4}$ ft., above which the walls are of red brick. The sash frames are of steel, the total sash area being approximately 40 per cent of the total wall space. The roof is carried on steel trusses with ample skylight area. The floors are of 4-in. concrete with $\frac{5}{8}$-in. mastic top.

7 The shop has three main divisions. The front one, facing the midway and occupying the entire front, is the machine section, consisting of two 100-ft. bays running parallel to the midway, the one adjoining the midway being $209\frac{1}{2}$ ft. long, while the inner bay is 182 ft. long. Each bay is served by a 10-ton electric crane of the open-latticework type of $96\frac{1}{4}$ ft. span, and a height to base of rail of $28\frac{1}{2}$ ft.

8 The freight-car erecting section is situated in the rear of the machine section, is 72 ft. wide and 405 ft. long, and is equipped with two 10-ton traveling cranes of $67\frac{1}{2}$ ft. span, and a height to base of rail of 27 ft.

9 The passenger-car erecting section consists of four bays of a total width of 110 ft. These four bays with the 72-ft. freight-car bay complete the full width of the back of the machine shop. Each bay of the passenger-car section is provided with a separate 2-ton traveling crane of 24 ft. 10 in. span and a height to base of rail of 21 ft.

10 Material is stored along the entire front of the shop, between it and the midway, in a yard 100 ft. wide served by a 10-ton crane of identical span and height with those in the machine section in anticipation of future shop extension. This crane runway extends beyond the shop limits and spans several tracks where cars of material can be readily switched and unloaded.

11 The arrangement of tracks in both freight- and passenger-

car erecting shops is shown in Fig. 1. Trucks are delivered from the car-truck shop by means of the track parallel to the south wall of the freight-car shop, and the turntables shown.

Crane Facilities

12 The whole area of the shop is traversed by electrically operated traveling cranes so arranged that it is possible to install an unusual number, and yet maintain for each crane complete freedom of operation at all times. The crane arrangement governed the shop layout. In order to make the machine section of the shop independent of the erecting sections so far as crane service was concerned, the crane runways in this section were installed in a direction transverse to those in the erecting shop. It was thus possible to equip the two machine bays with separate cranes, each having a wide range of action with no interference.

13 The machine layout is arranged to relieve the overhead cranes to the greatest possible extent, particularly in the handling of the larger members, such as center and side sills which must pass over two punching machines situated in the different bays. After the first operation the sills are transferred to the second machine by special devices independently of the overhead cranes. On completion of the machine operations they are skidded over to the assembling trestles in the erecting shop without assistance from the cranes.

14 The two cranes in the freight-car erecting shop, operating on the same runway, are entirely free from machine-shop handling, and as the first one is assigned to the preliminary assembly positions and the second to the final assembly positions, there is no overlapping or interference.

The Machine Equipment

15 The machine shop is equipped with the following machinery: Four automatic spacing punches, five coping punches, five high-speed punches, two horizontal punches, one 7-ft. 6-in. gate shear, one 10-ft. gate shear, one angle shear, one 36-ft. plate-edge planer, one 30-in. circular saw, one 30-in. metal band saw, one 7-ft. plate roll, one 10-ft. brake, one bulldozer, two special plate-drilling machines, and miscellaneous drill presses, all driven by independent motors. The arrangement of the machines is such that the material after each operation moves forward in the direction of the erecting shops, thereby reducing material handling to a minimum.

16 The high-speed punches are belt-driven direct from motor to flywheel, gears being dispensed with. Clutches are of the 6-point type. The heads are equipped with two punches which are controlled by gag levers. These machines are well adapted for punching the smaller plates for which metal templets are made up and into which holes are drilled; by inserting a pin or gage

in each successive hole and butting the plate against the pin, the desired spacing is obtained. For heavy punching four automatic spacing punches of moderate capacity were installed, and as the five additional coping punches are duplicates of those used in the automatic spacing tables, replacement can be made with but short delay.

The Machining Operations

17 The larger members such as sills, cover plates, side plates, etc., are stored in the yard in front of the shop. The smaller parts such as posts and braces, bolster and cross-bearer diaphragms, etc., are made by bulldozers and hydraulic presses in the blacksmith shop.

18 Narrow-gage service tracks carry material from the storage yard to the machines. The progress of material through the machine shop is as follows: Center sills and side-sill channels are loaded by overhead crane, brought into the shop on service lorries, and deposited on trestles opposite the traveler at the rear of center- and side-sill web spacing punch No. 775. Two air-operated traversing jacks lift the sills in pairs and place them on the traveler rollers ready to pass through the machine. On the far side of this punch, is an elevated runway carrying the traveler head which grips the sills with its projecting jaws and automatically spaces the punching. Along the traveler runway steel templets with projecting pins engage a trip lever suspended on the head of the traveler, close the electric circuit, arrest the travel of the head, and close the circuit of the punch control; the punch then completes the operation. After passing through this machine the sills are released from the jaws of the traveler, lifted by jib cranes and deposited back to back on the rollers of sill-flange spacing punch No. 777, which are placed directly opposite the delivery end of the web punch. On completion of the first operation the sills are disengaged from the head clamp and pushed back over the rollers to the starting point, where they are again turned over by a special device attached to the jib cranes and passed through the machine for the second operation on the opposite flanges. The sills are then lifted by the traveling crane of the inner section of the machine shop and placed on the rollers of coping punch No. 763 directly opposite, where draft-key and lever slots are punched, completing the operations on the sills.

19 The center-sill and body-bolster cover plates and similar plates are punched on spacing punch No. 750. The side plates are punched on automatic spacing punch No. 764 in a similar manner to the sills, passing through the first and second operations in pairs.

20 In all of these operations the passage through the machines is rapid and the accuracy of the spacing mechanism is such that the punching error is slight and far less than it would be were each hole marked off and punched independently.

21 Certain parts cannot be handled satisfactorily on the spacing

punches, yet the punching must be equally accurate or otherwise much of the benefit obtained from the spacing punches is lost when the parts are assembled. The machining of the ends of the side posts and braces is an example of this kind. In this case special combination dies shear the ends to shape and punch the group of holes in one operation.

22 The bolster cover plates and diaphragms are machined in a similar manner, the cover plates being passed over the spacing punch and the diaphragm flanges punched in one of the coping punches equipped with special dies for punching the flange holes in one operation. During these operations the machine is relieved of excessive load by varying the length of the punches, resulting in the holes being punched consecutively.

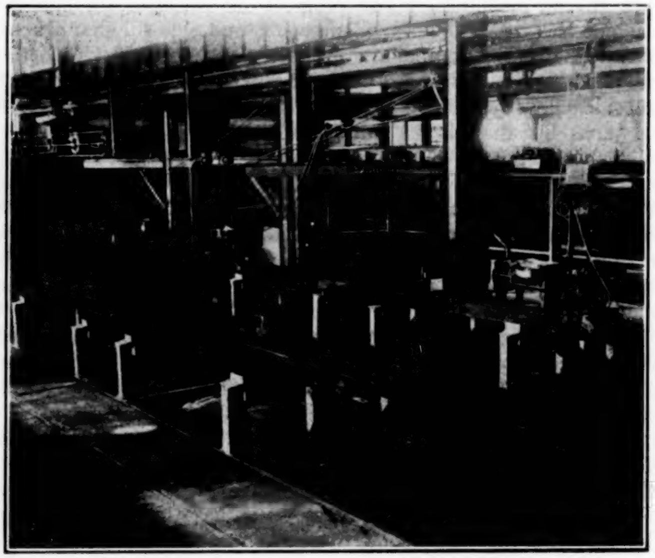


Fig. 2 Jigs Used in Assembling Underframes

23 The usual practice for freight-car work is to punch rivet holes to a diameter not exceeding that of the rivet, and, when the parts are bolted together, to ream the holes to a size not exceeding that of the rivet by more than $\frac{1}{16}$ in.

The Jig Method of Assembly

24 The erecting of steel-frame box cars by the jig method was originated at the Angus shops. By this method the underframe, side frames, and end frames are assembled on jigs as complete units ready for the final assembly of the car. The jigs consist of stands or

cradles by means of which the various members are accurately placed and held in proper relation to each other until they are riveted together. As each member lies flat in the jig the drawing together of the parts is reduced to a minimum, wedge bolts being used extensively for this purpose. A complete unit being assembled in one operation, the possibility of a cumulative error is avoided. The jigs dispense entirely with checking for squareness, alignment, and location of connection holes.

25 The center-sill channels after passing through slot-punching machine No. 763 are skidded from the idler rollers to trestles just inside the erecting shop, where the draft castings are temporarily bolted on and the holes reamed ready for riveting. An electric

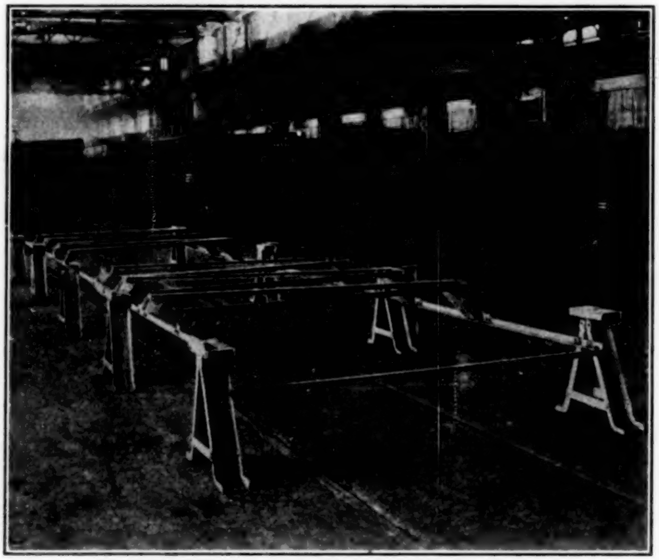


FIG. 3 JIG USED IN ASSEMBLING SIDE FRAMES

hoist on a runway below and clear of the overhead cranes, swings the sills into position and the draft castings are riveted on by a compression riveter. The individual sills are then moved across to a position on the left where the two center sills are assembled and riveted with bolster center castings and separators in position. The sills are placed on stands on which are four fixed pins corresponding to four rivet holes in the sills at the center line of the bolsters. Perfect alignment of the two sills is thus assured, which simplifies the application of cover plates later on. The draft gear is also applied in this position.

26 The next step is the assembling of the underframes, and as this is a lengthy operation, four positions are assigned for the

purpose as indicated in Fig. 1. These jigs, as shown in Fig. 2, consist of four steel cradles located at bolster and cross-bearer centers, and so arranged that each member of the underframe is held in proper alignment and at centers that will coincide exactly with connections on the side frames. The bottom cover plates of the bolster and cross-bearers are first placed on centering pins; the overhead crane then places the center sills in position, to which are attached the bolster and cross-bearer diaphragms with their cover plates and center-sill cover plate complete, after which the assembly is bolted together and the holes reamed ready for riveting.

27 The underframes are then swung over by the overhead crane

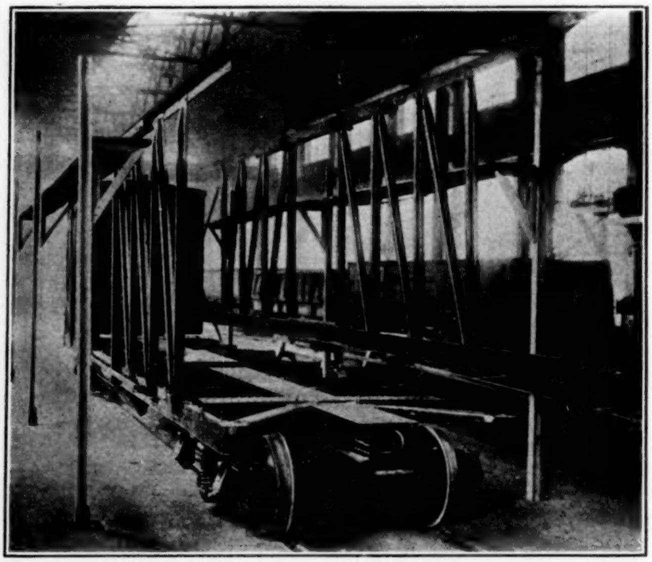


Fig. 4 Placing the Side Frames in Position

to the riveting jigs on the right, which are constructed similar to those used for assembling, thereby maintaining the proper alignment. Each riveting position is equipped with two 50-ton compression riveters suspended from swinging jib cranes, the posts of which are in line with columns to the right. The crane jibs are 21 ft. long, with runways for air hoists. The suspension mechanism is so arranged that the riveters can be tilted to drive the rivets in the inclined bottom flanges of the bolster and cross-bearers. The compression riveters were specially designed on the scissors principle, with a thin nose to permit of the top and bottom rows of rivets being driven without turning the underframe over.

28 The side frames, consisting of the side sill, side plate, post,

braces, door posts, and track, are assembled as a unit on a jig frame situated abreast of the underframe jigs. This jig, as shown in Fig. 3, consists of channel-iron stands, the four corner ones of which are capped with short sections of channel iron in which are holes for locating exactly the side sills and plates. The stands on either side are tied together with angle bars which carry additional cross-bars upon which the various members of the side frames are placed in proper relation to each other. In this position the side frame is temporarily bolted, reamed, and riveted ready for the final assembly.

29 Originally the end frames were assembled on jigs similar to those used for the side frames, but in recent years the end-post

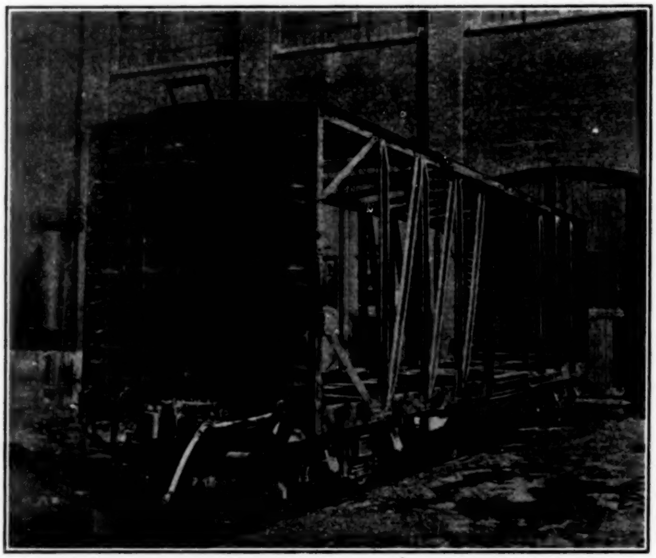


Fig. 5 Car with Steelwork Completed and Ready for Finishing

construction has been replaced by corrugated-steel ends. These steel ends and the end sills are assembled on trestles located between the final assembling tracks, and when temporarily bolted together they are skidded to the second and third positions where they are then reamed and riveted. In the fourth position end ladders, roof-frame brackets, and other parts are applied, the completed ends being placed opposite the final assembling position.

The Final Erecting

30 As the trucks are assembled and painted, they are delivered from the truck shop and enter the erecting shop by the side door

immediately ahead of the underframe and side-frame jigs, where they are turned on a turntable and placed in position on the assembling track. The underframe completed in the riveting jig is then lifted by the overhead crane and placed on the trucks, the slings are released, and a steel end is next placed in position and bolted on the end farthest away from the side-frame jigs; the side frames are then placed in position as in Fig. 4, and finally the second end, along with the center and side-sill cross-ties. The brake-cylinder reservoir and piping are also applied at this time.

31 The car is then moved by car haul to the second position where the assembled members and the roof framing are riveted in

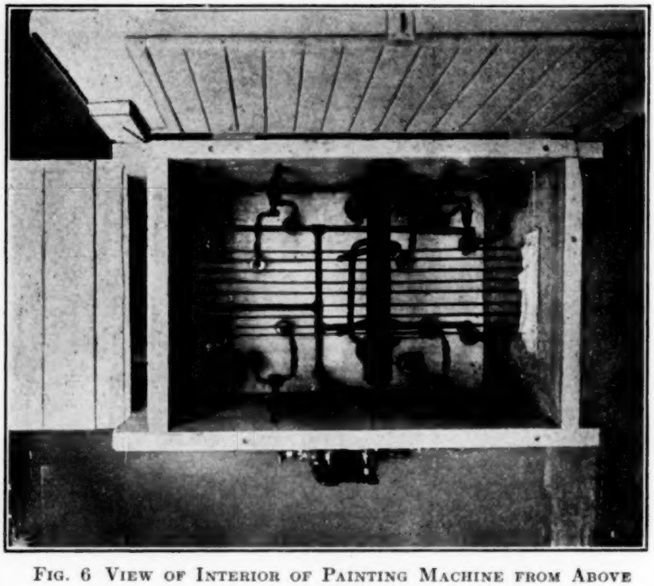


Fig. 6 View of Interior of Painting Machine from Above

place. In the third position the safety appliances, brake rigging, couplers, uncoupling rods, etc. are applied, the remainder of the riveting completed, and the entire frame as shown in Fig. 5 is then sprayed with the priming coat of paint ready for finishing.

Finishing the Car

32 The steel frames are switched each day to the wood-freight-car shop where the decking, sheathing, roofing, and doors are applied and the painting operations are completed. The decking is of $1\frac{3}{4}$-in. red pine and the sheathing of $1\frac{1}{2}$-in. Douglas fir, both having tongued-and-grooved joints. The lumber is kiln

dried, the moisture content being carefully limited in order to prevent the possibility of further shrinkage later on.

33 Before leaving the planing mill the sheathing, roofing, and running boards receive their priming coat of paint in a painting machine recently developed at the Angus shops and which differs from those in use elsewhere. The boards on leaving the matcher pass automatically through the painting machine shown in Fig. 6 where they are sprayed by a series of nozzles which can be set in any desired position according to the surfaces required to be painted. The paint is drawn up through suction pipes from the bottom of the box by means of air jets blowing across the nozzles, and as ejected it is atomized by the air and blown on to the boards in the form of a fine spray. The amount of paint to be applied is controlled by air valves or by regulating the speed at which the boards pass through the machine. No brushing or wiping is necessary. The boards on leaving the machine are piled on trailer trucks and distributed by tractors to the shops when dry. These machines will paint at the rate of 200 running feet per minute, which is about as fast as the boards can be conveniently piled for drying.

34 The first operation in the wood-freight-car shop is to apply the decking, the joints of which are previously coated with a thick paint compound, as are also the ends of the boards making contact with the bottom boards of the side sheathing previously applied.

35 The side sheathing which has already received the priming coat of paint is next applied, and to insure that the sides of the car will be watertight, the joints are coated with paint compound, after which they are wedged down into position and bolted to the framing. The end lining is then applied in a similar manner but vertically. In the succeeding operation the roof is applied, the boards and metal sheets of which have previously been primed.

36 After the doors are hung and the remainder of the safety appliances have been installed, the car is given two additional coats of paint and stenciled. It is then complete and ready for service.

DISCUSSION

F. O. Whitcomb.[1] The construction of steel cars in railway shops presents some differences from contract-shop practice. Since a contract shop seldom, if ever, puts through two lots of cars which are identical, and further, speaking for Canada, since there are never any very large orders executed at one time, it is unwise to spend much money on large or even medium-sized special tools, such as blanking and gang tools. These do not represent a capital expenditure, and, naturally, must be charged to the individual order for which they are designed. Therefore our practice is to use the spacing punches, multiple punches of individual set-up, and single-

[1] Production Engineer, Canadian Car & Foundry Co., Ltd., Montreal, Canada.

punch work from templets. We insist on accurate work, and when the car is constructed we find little difficulty in assembling it square and true to dimensions. The railway shop man can then work with the car designer and build the car around the existing tools.

Construction. In Par. 26 the author describes at length the jig method of building underframes of freight cars. We disagree with him on this practice. With accurate punching and pressing of members an underframe can be built on the skid method very accurately, and with much greater despatch, the result being a cheaper car. The first operation of boxing sills on the skids starts a progressive system of building, and any desired production up to the limit of the track may be obtained by adding fitting and riveting gangs. Our practice is to fit and assemble the underframe in six positions on the skids, and at this point the underframe is placed on its own truck. In ten moves on the track the car is ready for the woodwork, having received its priming coat of paint on the steelwork. With such an arrangement an average of 42 cars per day has been made, taken over one month, with a production of 52 cars in 11 hours as the maximum for a day's work. For passenger work and street cars it is absolutely necessary to build the underframe in jigs in order to put in the camber of the sills and level the floor beams.

Electric rivet heaters are valuable in Canada where electric power is cheap and oil expensive. In our plants these machines have proved to be very economical and remarkably free from maintenance. Another great advantage is the saving of rivets, which is very marked, as a burnt rivet is practically unknown. On a 9-hr. continuous run a three-head machine showed a saving of $8.83, while a five-head machine showed a saving of $12.65 for the same time.

The continuous painting machine is to be commended, as should be any other practice which will make for the longer life of cars. The railroads would do well to inaugurate an intensive campaign for better paint conditions.

In applying the lumber to our cars we follow the same method as for building the steel part of the cars, viz.: the track system, the flooring, sheathing, and roofing, and two coats of paint being put on in this operation. After this the cars leave the track and are set off to have air brake and stenciling applied, and are put through the shipping track inspection prior to shipping.

Max Toltz. When we first designed steel cars the main point was to reduce the dead weight of the car as compared with the live weight. In recent years the dead weight has grown in proportion so that it now is 45 per cent of the live load. Care on the part of the car designers in devising new methods of reducing the dead weight of a car would probably be of considerable benefit. The use of the proper material, such as nickel steel, etc., would effect a considerable reduction. It is not the cost of the car but its

weight that should be reduced. The less weight we carry on the rails, the more money we can make.

Augustus Smith. The New York Central have, within recent years, adopted a container system so that freight can be put in boxes and lifted bodily on and off the car frames. It is to be hoped that the box-car designers will design box cars so that the top can come off to enable a crane to get material which is too heavy to roll through the door, or to lower trays down through the roof.

The increasing difficulty of obtaining manual labor makes it desirable to so design box cars that they will lend themselves more readily to the handling of their contents by mechanical means.

The Author. Mr. Whitcomb questions the advisability of constructing special blanking and punching tools for other than very large orders, such never being executed in Canada. He apparently overlooks the fact that Canada has the two largest railroads on this continent, and has a number of railroad and contract car-building plants capable of turning out any number and variety of cars which may be required. Also these railroads have a tendency to standardize the designs of their cars and to repeat their orders for such designs, so that both railroad and contract shops will find it economical in many cases to make special shop equipment and tools.

The author is rather of the opinion that hesitancy to construct such tools with the intention of saving in tool expenditure unnecessarily increases final production costs. It has been found that the cost of preparing such tools, even for orders of 500 or 1000 cars, proved to be a good investment. By using tools of this description "marking off" costs are eliminated and the various parts of the car are more readily and cheaply assembled, owing to the parts being more accurately machined. Mr. Whitcomb also favors the skid method for building underframes, except for passenger or street cars, for which he considers the jig method essential in order to obtain the necessary camber and floor-beam alignment. Even for freight cars it is not good practice to depend entirely on the accuracy of the punching and presswork when assembling an underframe 40 ft. long, or more, for unless the bolsters and cross-bearers are in line one with the other, a twisted underframe will result. Furthermore the whole superstructure will later be stressed to take this same irregular formation.

Aside from the question of production and costs as between these two methods, would it not appear of greater advantage to build large orders of freight cars on jigs than relatively small orders of passenger cars? As regards production, the jig as compared with the skid method reduces the number of moves or operations, for when once the center sills are riveted, the underframe is assembled and riveted complete in two moves ready for the application of the superstructure. The steel underframe of a box car having channel

centers sills, when assembled on jigs is completed ready for the woodwork in nine moves, and similarly the fish-belly type of construction is completed in seven moves, which is less than that claimed for the skid system. A further and very important advantage gained by the adoption of the jig system is that it prevents overcrowding. For example, the assembling and riveting of the underframes can be divided into three or more batteries, each battery having three assembling jigs and one riveting jig, making in all twelve jigs. Every one of these can be fully manned without interference or congestion, which will have an important bearing on production costs.

No. 1896

EFFECT OF VARIATIONS IN DESIGN OF MILLING CUTTERS ON POWER REQUIREMENTS AND CAPACITY

By Benjamin P. Graves,[1] Providence, R. I.

and

James A. Hall,[2] Providence, R. I.

Members of the Society

A series of tests to determine the effect of changes in the number of teeth, spiral angle, rake angle, and cutting speed of the cutter on the power consumption, tendency to chatter, and stresses set up in the machine, led to the following conclusions:

Coarse-tooth milling cutters require less power to remove a given amount of metal per minute than fine-tooth cutters.

When compared on a chip-per-tooth basis, the finish given by a coarse-tooth cutter is better than by the fine-tooth cutter.

Low-cutting-speed operation of fine-tooth cutters to give a large feed per tooth increases the stresses in machine and fixtures.

Fine-tooth cutters are inferior to coarse-tooth cutters when the relative tendency to chatter is considered.

Moderate rake angles reduce the power consumption and are desirable on all cutters to be used on mild steel. Large rake angles increase the tendency to chatter.

The spiral angle has little effect on power consumption; a considerable spiral angle is desirable, however, as fewer teeth may be used, smoother action is obtained, and the tendency to chatter is reduced.

With certain exceptions noted in the paper, coarse-tooth cutters of the type now generally manufactured are superior to fine-tooth cutters for all classes of production work.

A SERIES of investigations at the plant of the Brown & Sharpe Mfg. Co. to determine the effect of changes in the design of milling cutters form the subject of this paper, which comprises a study of the effect of changes in the number of teeth, spiral angle,

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[2] Associate Professor of Mechanical Engineering, Brown University.

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rake angle, and cutting speed of the cutter on power consumption, tendency to chatter, and stresses set up in the machine.

2 As cutters and not machines were to be compared, variables due to the machine alone should be eliminated. It therefore seemed best to make most of the tests with the work held rigidly on the milling-machine table and to determine the power used by the cutter from the figures of the input into the motors driving the machine.

3 An old-style Brown & Sharpe No. 3 heavy plain milling machine with a constant-speed drive shaft was used. Upon its table a fixture was placed into which dovetailed steel specimens 6 in. wide and 18 in. long could be fastened by means of a gib. The cutters were also rigidly held as they were keyed on a $1\frac{1}{2}$-in. arbor, arbor supports were used on both sides of the cutter, and the outside braces were bolted during all of the tests. Errors due to distortion or vibration in the machine or arbor were thus eliminated as far as possible.

Measurement of Power Used by Cutter

4 The power figures desired in making comparisons between cutters are the amounts required by the cutters alone and should not include the friction losses in the machine unless the friction has some direct relation to the pressure exerted or power required by the cutter.

5 The spindle and table were driven by separate motors. The efficiency of the spindle drive at different capacities and speeds was determined by a prony brake on the arbor. A horsepower-kilowatt curve was then drawn with the power delivered to the arbor, and the difference between the no-load kilowatt reading and the reading under load as coördinates.

6 The proper treatment of the power required by the table is a more difficult problem. Here the no-load kilowatt reading includes the motor losses and the friction losses in all of the bearings up to the table clutch, and these losses remain nearly constant for all loads. The additional power required when the table is operating under load is not only that needed to overcome the pressure of the cutter, but a much greater amount due to the friction loss in the lead screw and thrust bearing caused by that pressure. As this friction varies with the pressure, it seems obvious that it should be charged against the cutter if conditions calling for different pressures are to be compared. The authors therefore took as the net power required by the table the difference between the motor output under load and that when driving the feed mechanism alone.

7 Any milling-machine table drive is necessarily of low efficiency due to the fact that the table must not continue to move and drive the lead screw after the table clutch is thrown out. This limits the efficiency of the drive to a reasonable margin of safety below 50 per cent when operating on a high-speed return,

and to a much lower figure at cutting speeds. This low efficiency makes little difference under ordinary conditions as a very small proportion of the total power of the machine is taken by the table. This would be changed if the practice of using low cutting speeds were carried to an extreme. This condition is further discussed later.

Cutters

8 High-speed-steel cutters $3\frac{1}{2}$ in. in diameter were used in all of the tests, excepting a few tests with two 6-in. side milling cutters. All of the $3\frac{1}{2}$-in. cutters used in the tests to determine the effect of the number of teeth had 30-deg. spiral angles and 10-deg. rake angles. As the formula derived by Professor Airey and Mr. Oxford[1] called for 20 teeth in such a cutter, this was put at one end of the series, a Brown & Sharpe standard cutter with eight teeth came in the middle, and a four-tooth cutter was added to give an extreme condition of coarse spacing. A number of tests were also made with a 10-tooth cutter. Additional comparisons were made between the two 6-in. side milling-cutters, one of which had 12 teeth and the other 26 teeth. The latter cutter was $^{15}/_{16}$ in. wide and the former 1 in.; otherwise they were exactly alike.

9 Other sets of cutters alike in every particular except the element to be studied were used in the tests to determine the effect of variations in spiral angle and rake angle. The sizes of these cutters are given in the discussion of the results.

Metal Used in the Tests

10 As the only object of the investigation was to compare cutters, mild steel was used in all the tests. This was done to enable the authors to make enough cuts to be sure of their results and to eliminate as far as possible variations due to lack of uniformity in the metal being cut. Fortunately most of the tests were made on blocks which were fairly uniform.

11 The properties of the steel in all the specimens used in the tests were about the same with the exception of the $\frac{1}{2}$-in. bar. The values for tests on specimens 0.505 in. in diameter and 2 in. long are given below:

	$^1/_2$-in. Bar	Other Specimens
Elastic limit, lb. per sq. in.	50,300	42,000
Breaking load, lb. per sq. in.	73,500	68,000
Elongation, per cent	34	35
Reduction of area, per cent	62	62
Brinell hardness	126	126

Method of Making Tests

12 A light cut was always taken along the specimen before starting any test, to guarantee that the depth of cut would be constant. The table was then fed up the desired amount, and

[1] The Art of Milling, Trans. A.S.M.E., vol. 43 (1921), p. 549.

from two to six tests were made at this depth of cut in the 18 in. of length of the block. At the end of the first cut the table was run back and the depth of cut checked. The exact figure for the feed was determined by taking the time with a stop watch and measuring roughly the distance traveled with a steel scale and

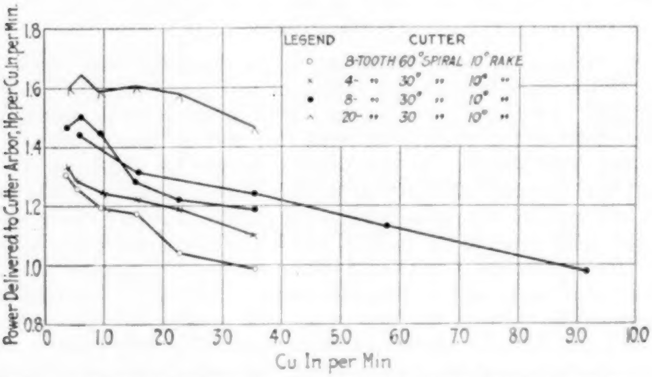

FIG. 1 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT DEPTH OF CUT AND VARIABLE FEED

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.100 in.; cutting speed, 94 ft. per min.)

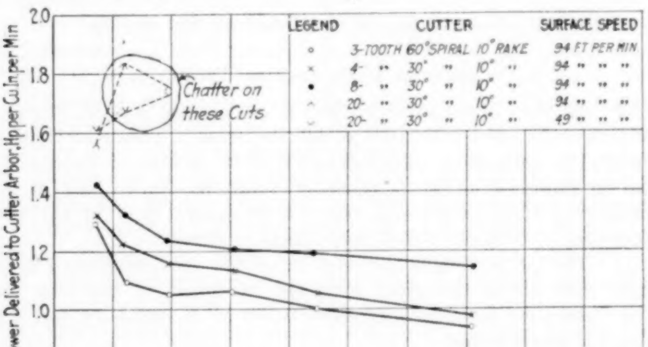

FIG. 2 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT SPEED, CONSTANT DEPTH OF CUT, AND VARIABLE FEED

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.200 in.)

correcting this figure to 0.001 in. by reading the dial on the lead-screw handwheel. The exact amount of metal removed per minute was thus determined.

13. Before and after each test, readings were taken to deter-

mine the friction load, and several readings were made during the test to determine if the power remained constant. The differences between the readings under load and the no-load figures multiplied by the proper constants gave the horsepower delivered to the cutter arbor and to the table lead screw. A number of tests

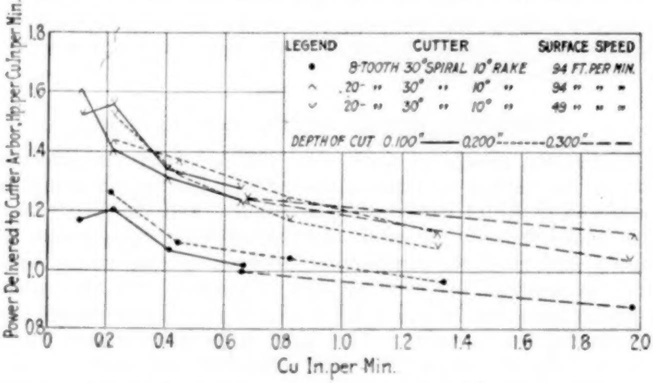

FIG. 3 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT SPEED, VARIABLE FEED

(Material cut, mild steel; width of cut, 1.012 in.; depth of cut, 0.100 in. to 0.300 in. as indicated in figure.)

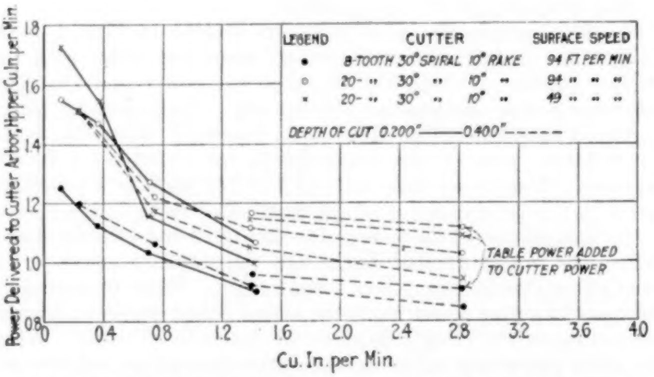

FIG. 4 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT SPEED, VARIABLE FEED

(Material cut, mild steel; width of cut, 0.535 in.; depth of cut, 0.200 in. and 0.400 in. as indicated in figure.)

were also made on a special fixture to measure horizontal and vertical pressures by use of two Kenerson traction dynamometers.

14 The results are plotted in the form of curves having as one coördinate either horsepower per cubic inch per minute or pressure along the table. The former figure is used instead of the more

common cubic inches per horsepower, as in some cases the sum of the cutter and table power is given as well as the cutter power alone. For the other coördinate cubic inches per minute, feed per tooth, or maximum chip thickness is used. The latter quantity was used extensively in the paper by Professor Airey and Mr. Oxford, and they claim that the power required per cubic inch of metal per minute will be the same for the same maximum chip thickness for all cutters having similarly shaped teeth, irrespective of the number of teeth.

Effect of Number of Teeth in Cutter

15 Several groups of tests were run to study the effect of varying the number of teeth in the cutter. In the first group the speed and depth of cut were kept constant and the feed varied. This varied the chip per tooth and the output in cubic inches per minute and gave the comparison between cutters where equal finish is required for corresponding capacities. In the second group of tests the feed and depth of cut were kept constant and the cutting speed varied. This gave a constant output and a variable chip per tooth. For the same feed per tooth the best surface finish is secured with the coarsest-tooth cutter. In the third group the depth of cut was kept constant and the cutting speed varied with the feed so as to give a constant chip per tooth. The output varied with the speed. This showed the effect of the cutting speed on power requirements.

16 The tests in the first group are plotted in Figs. 1 to 4. These show that for the same cutting speed the cutter with the smaller number of teeth always takes less power and that this difference is more noticeable on wide cuts. The fine-tooth cutter chattered considerably on these cuts, especially when the depth was 0.200 in., where it was impossible to use it beyond a limited capacity. This shows that the width of the work is a necessary factor in the determination of the number of teeth in the cutter.

17 In a number of the tests plotted in Figs. 3 and 4 the fine-tooth cutter was run at approximately half the standard speed to see what effect this had on power consumption. When compared on the basis of cutter power alone the slower speed showed a saving, but not enough to bring it down to the coarse-tooth cutter. When the table power was added to the cutter power, the relative advantage of the slower cutting speed practically disappeared.

18 The real criterion of the effect of chip size on power requirements can best be determined from tests with constant output but variable feed per tooth. The second group of tests were along these lines. See Figs. 5 and 6. These show that the power efficiency is greater with coarser-tooth cutters even when compared on a chip-per-tooth basis and when only the power required by the cutter is considered. If table power is added to cutter power the advantage of the coarse-tooth cutter is even more

apparent. In these curves the sum of the cutter and table power comes to a minimum as the cutting speed is reduced, after which the table power increases more rapidly than the cutter power is reduced by the increased size of the chips. This point is referred to later.

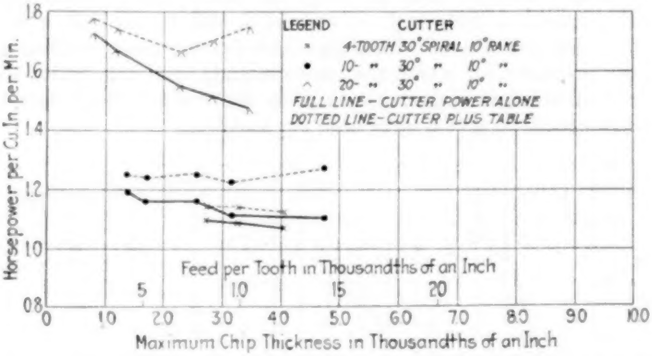

FIG. 5 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT FEED, CONSTANT DEPTH OF CUT, AND VARIABLE SPEED

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.100 in.; feed, 4.1 in. per min.)

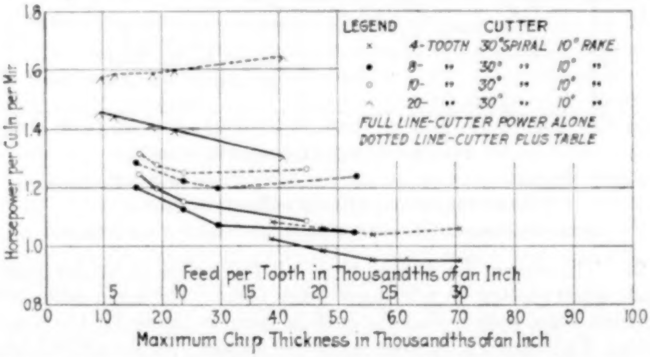

FIG. 6 POWER REQUIRED BY COARSE- AND FINE-TOOTH CUTTERS — CONSTANT FEED, CONSTANT DEPTH OF CUT, AND VARIABLE SPEED

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.050 in.; feed, 8.20 n. per min.)

19 Fig. 7 gives the results of a series of tests where the horizontal thrust along the table was measured at several different cutting speeds and two different outputs for each of five cutters. As the depth of cut was small, the power required by the cutter is practically equal to the product of the pressure multiplied by the velocity of the cutter. This checked rather closely with the amount determined from the wattmeter readings where these were taken.

20 In this figure note that the pressure increases as the cutting speed is decreased to give a larger feed per tooth. Also if two cutters with different numbers of teeth are operated at the same total feed and also the same feed per tooth, the pressure caused by the cutter with the larger number of teeth is greater by a somewhat larger amount than given by the tooth ratio. The eight-tooth cutter is a little high on this basis, due to the fact it was duller than the others. Machines and fixtures have limitations in the pressures for which they are designed, and this limits the extent to which it is desirable to go in running fine-tooth cutters at slow speeds to give a large chip per tooth.

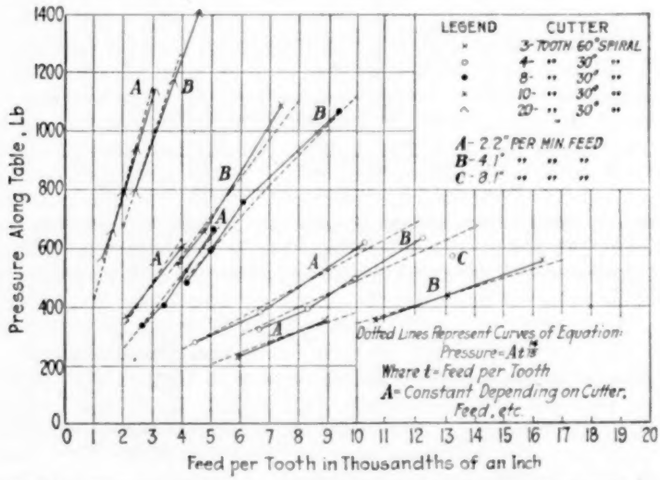

FIG. 7 EFFECT OF NUMBER OF TEETH IN CUTTER ON HORIZONTAL PRESSURE ALONG MILLING-MACHINE TABLE

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.050 in.)

21 The curves for higher feeds all fall below those for lower feeds when plotted on a feed-per-tooth basis. The main difference between these tests is that the cutting speed is higher for the larger feeds. This raised a question as to the accuracy of the assumption often made that the cutting speed has no effect on the energy required per cubic inch of metal removed, and a special series of tests were made to study the effect of this variation. The results of these tests are presented a little later in the paper.

22 The data given in Fig. 7 were plotted on logarithmic cross-section paper and found to form a family of curves of the equation:

$$P = At^{14/15}$$

where P = horizontal pressure along table
t = feed per tooth, and
A = a constant depending on cutter and feed.

The number of tests involved is not sufficient to commit the authors absolutely to the power "$^{14}/_{15}$" but they feel sure that the correct figure is only a small amount below unity.

23 Having an equation in the above form, it is possible to calculate for shallow cuts the proper relation of cutting speed and table feed to give the minimum power noted in the discussion of Figs. 5 and 6. Given

V = cutting velocity in feet per minute
v = table velocity in feet per minute
E = efficiency of the table

$$\text{Horsepower} = \frac{P}{33000}\left(V + \frac{100v}{E}\right) \quad . \; . \; . \; . \; . \; . \; [1]$$

But $P = At^x$ and $t = Bv/V$, where B = a constant depending on the cutter. Substituting these values in [**1**] and solving for the value of V to make the power a minimum for any given feed and table efficiency,

$$V = \frac{100\,x}{(1-x)E}v \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [2]$$

For example, if $x = {}^{14}/_{15}$ and E = 14 per cent, $V = 100v$.

24 Equation [**2**] gives a limit beyond which there is no saving in power when the cutting speed is lowered to give coarser chips per tooth.

25 Figs. 8 and 9 give the results of studies with the side milling cutters. Fig. 8 shows tests where the cutting speeds were such that the fine-tooth cutter took practically the same chip per tooth as the coarse-tooth on corresponding cuts. Compared on the basis of cutter power alone, the coarse-tooth cutter was superior; this advantage became still more pronounced when the table power was added. In attempting to carry the cut of 4 cu. in. per min. with the fine-tooth cutter, the key in the arbor sheared.

26 Other series of tests with these cutters are shown in Fig. 9, the comparison being on a maximum-chip-thickness basis. Here again the fine-tooth cutter required more power and also proved inferior from the standpoint of chatter.

27 Frequently in comparing results on a feed-per-tooth or a maximum-chip-thickness basis the power required per cubic inch per minute and the pressures along the table have been less for higher cutting speeds. The tests shown in Fig. 10 were therefore run to check up this relation. The chip per tooth was kept nearly constant as the cutting speed was increased. The results show that the horizontal pressure and the power required per cubic inch of metal per minute decreased as the cutting speed increased. The vertical pressure between cutter and work remained constant.

28 It would doubtless seem from the foregoing that the efficiency of milling is increased by greater feeds per tooth. If these heavier chips are secured by using lower cutting speeds, the loss in the table becomes greater, the cutter itself requires more power

per unit of metal than would be required for the same chip at higher speed, and the pressures between cutter and work become greater. The answer from the standpoint of power efficiency alone is then to use the cutter with the fewest possible teeth and to

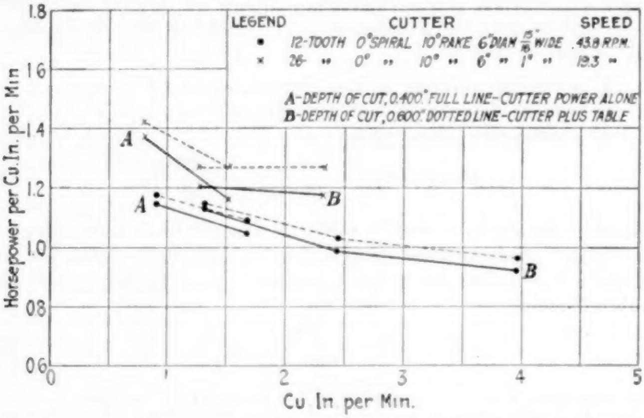

FIG. 8 POWER REQUIRED BY COARSE- AND FINE-TOOTH SIDE MILLING CUTTERS CUTTING MILD STEEL — CONSTANT SPEED, VARIABLE FEED

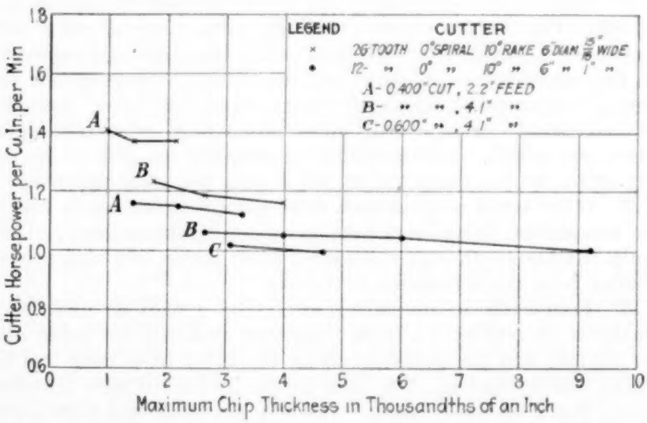

FIG. 9 POWER REQUIRED BY COARSE- AND FINE-TOOTH SIDE MILLING CUTTERS CUTTING MILD STEEL — CONSTANT SPEED, VARIABLE FEED

run it at higher speeds where the size of the chip is too great. The limitations to this practice are the danger of hammer or shock due to lack of continuity of cutting and the possibility of overheating the edges of the teeth due to the heavy pressures and high speeds. The structural strength of the tooth itself is not a limitation in

well-designed cutters as this is practically always greater than the strength of the key or the arbor. The relation between the overheating of the edges of the teeth and the number of teeth in the cutter can only be settled by judgment, as no work has been done on the relative effect of cutting speed, volume and length of chip, ductility of the metal, and energy required per chip on rise in temperature.

Effect of Spiral Angle

29 A large number of runs were made on a set of three 10-tooth cutters with spiral angles of 10, 20, and 30 deg., respectively, and

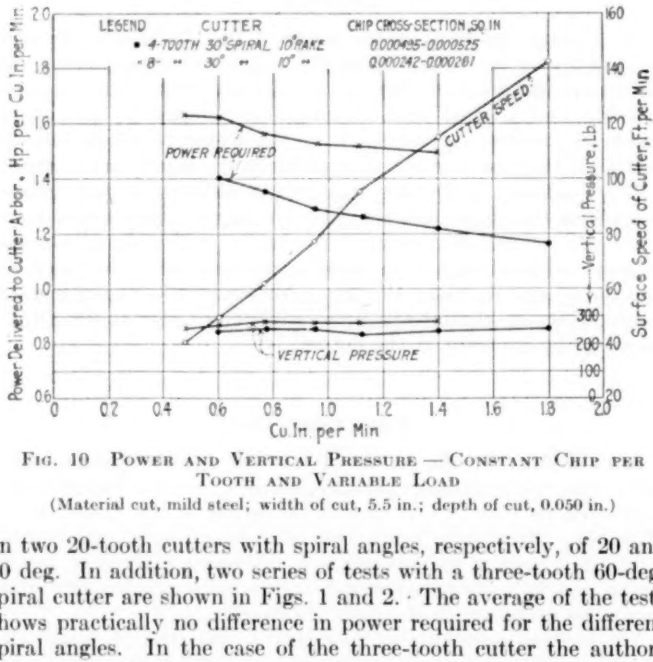

Fig. 10 Power and Vertical Pressure — Constant Chip per Tooth and Variable Load

(Material cut, mild steel; width of cut, 5.5 in.; depth of cut, 0.050 in.)

on two 20-tooth cutters with spiral angles, respectively, of 20 and 30 deg. In addition, two series of tests with a three-tooth 60-deg. spiral cutter are shown in Figs. 1 and 2. The average of the tests shows practically no difference in power required for the different spiral angles. In the case of the three-tooth cutter the authors believe that the lower power requirement is due to the number of teeth instead of the spiral angle.

30 While increased spiral angles showed no advantage from the standpoint of power, they were of very great benefit in reducing tendency to chatter and in giving smoother action. The vertical pressure between cutter and work was also somewhat less. The end thrust with the 60-deg. spiral cutter was not noticeable as far as the machine was concerned.

Effect of Rake Angle

31 Tests were made with three 10-tooth cutters having rake angles of 0, 10, and 20 deg. and with two 20-tooth cutters having

0 and 10 deg. rake. The increase in rake angle from 0 to 10 deg. caused a saving in power of 20 to 25 per cent, with a somewhat smaller saving on the increase to 20 deg. Tendency to chatter increased only slightly between 0 and 10 deg. rake, but became so great as to limit definitely the capacity of the cutter when increased to 20 deg. This limitation seemed to be more serious than the danger of burning or chipping the edge. Rake angles of 10 deg., therefore, are desirable on cutters, but any considerable increase beyond this point limits the usefulness of the cutter.

Chatter

32 Chatter rather than the power of the machine is often the limiting factor in milling. For this reason many engineers urge that more rigid machines be built to eliminate this trouble, without realizing that a great deal may be accomplished by the proper choice of cutters. A summary of the elements which affected chatter in these tests is therefore given in the following paragraph.

33 Tendency to chatter was greatly reduced as the number of teeth in the cutter approached a low figure. Increase in spiral angle also reduces the tendency to chatter, and a combination of wide-spaced teeth and steep spiral angle in a cutter will give the maximum capacity along this line. Moderate rake angles increase the tendency to chatter but little, while large rake angles greatly decrease the capacity of the cutter.

DISCUSSION

John Airey. In the matter of rake the authors agree substantially with the recommendations made as a result of experiments [1] conducted by Carl J. Oxford and the writer at the University of Michigan. Practically every one who looks into the subject becomes convinced of the advantage of increasing rake up to certain limits, and it is inexplicable that such a large number of cutters should be marketed as standard with zero rake.

The authors state that as speed is increased the efficiency with which metal can be removed is decreased. While this is contrary to the results of his experiments, the writer is not prepared to disagree with it. In his chip-investigator work [1] the average of some 1200 experiments, *at three different speeds*, showed a maximum variation in efficiency of three per cent. As the region of speed then investigated was considerably lower than that dealt with by authors, the two results can be reconciled.

The method used for measuring power is of fundamental importance, and the writer takes exception to that used by the authors. The same method was fully considered and rejected in

[1] On the Art of Milling, by John Airey and Carl J. Oxford, Trans. A.S.M.E., vol. 43 (1921), p. 549.

outlining the Michigan experiments. In this method the measuring instruments have to be calibrated; the motors have to be examined for frictional losses; the machine has to be similarly examined; as also the feed mechanism; and in each one of these an error enters.

There is a much more important objection, however, than these errors. The frictional loss in the machine is not the same during cutting as it is when the machine is running under a prony-brake test, and the additional friction caused by the end thrust of the cutter should not be charged to the cutter. In investigating the fundamental action of a cutter it is necessary to distinguish between energy absorbed by the machine and energy transformed into heat at the cutting edge. The former does not affect the grinding life of the cutter. The latter certainly does. While machine loss strictly traceable to the cutter should be investigated and considered, this should not be confused with energy absorbed by the cutter proper. The obvious place to measure the energy is right at the cutter, where it is necessary only to measure the torque exerted and roughly measure the speed of the machine, because the relation of the feed to the spindle revolutions is known, irrespective of speed.

The authors state that coarse-tooth cutters always show up advantageously from the standpoint of finish. There are two points of view to be considered: First, if the cut is solely a finishing cut and the same feed per revolution be given to both coarse- and fine-tooth cutters, then admittedly the metal is removed less efficiently with the fine-tooth cutter, due to the decreased chip thickness. However, the energy actually used in metal removing proper is a negligibly small proportion of the total energy used in finishing cuts.

Second, if the cut is a roughing cut and must also produce a finished surface, it is generally argued that revolution marks indicate the degree of roughness of surface. The pitch of the revolution marks is feed per revolution, and it will therefore be proportionate to number of teeth in cutter for a given chip thickness.

While revolution marks leave a surface that is a series of waves, the inexactitude of a supposedly plane surface is measured by its maximum deviation from a true plane rather than by the period of cyclical change. For a given cutter diameter and feed per revolution there is a certain critical eccentricity of cutter relative to spindle such that if the actual eccentricity is less than this critical value, any added feed per revolution merely increases the pitch of the wave but leaves the depth constant. This is determined by the maximum deviation from concentricity of cutter.

Table 1 enables users of milling machines to judge roughness for any specific set-up. For a given cutter diameter and feed per revolution, if the cutter runs as true as, or more nearly true than, the value given in the table, any increase in feed per revolution

will only increase the period of wave, leaving the maximum deviation from a true plane unaffected.

From the foregoing it is conceded that in the majority of cases the depth of wave will be increased with an increase in the number of teeth. But this increase of depth is not continuous as spacing is reduced because the limiting condition, at which depth of wave remains stationary for subsequent increases of feed per revolution, is never far away with a well-set-up job.

Tooth Spacing. As a result of their experiments, Mr. Oxford and the writer arrived at the conclusion that for a given material and shape of tooth the sole criterion of the efficiency with which metal can be removed is the chip weight, or, to be more exact, the maximum chip thickness. The spacing of the teeth or the diameter of the cutter has nothing to do with this, for the following reasons:

Consider a planing machine with two mathematically identical tools of certain definite rake and clearance angles, depth of cut, feed, etc., situated in line, say, 12 in. apart, and taking precisely the same cut in the same material. The same resultant force will

TABLE 1 CRITERIA FOR DETERMINATION OF ROUGHNESS OF MILLED SURFACES

Diameter of cutter, in.	Feed per revolution, in.	Deviation of cutter from true concentricity, in.	Diameter of cutter, in.	Feed per revolution, in.	Deviation of cutter from true concentricity, in.
3	0.1	0.0008	6	0.2	0.0017
3	0.2	0.0033	6	0.3	0.0037
4	0.1	0.0006	9	0.2	0.0011
4	0.2	0.0025	9	0.3	0.0025
5	0.1	0.0005	12	0.3	0.0019
5	0.2	0.0020	12	0.4	0.0033
5	0.3	0.0045			

be found acting on each of these two tools. If this distance of 12 in. is gradually decreased, eventually the two tools will be so close that the free-cutting action of the second tool will suffer by its proximity to the first tool, and we naturally would expect a lack of equality in the resultant forces. To summarize: The two cutting edges will take the same cuts and exactly equal forces if they are so far apart as not to interfere with each other. The same reasoning should apply to a milling cutter.

The second reason is the result of the Michigan experiments. Various tests on the chip-per-tooth basis were compared and although the number of teeth in the cutters varied, all the points fell on one even curve, thus proving that the chip thickness is the sole criterion of the efficiency, assuming the shape of an individual tooth to be the same in all cases.

The third reason is based on the experiments made by Mr. A. L. De Leeuw and described in a paper presented before the Society in 1911.[1] His experiments, when compared on a chip-per-tooth basis,

[1] Milling Cutters and Their Efficiency, Trans. A.S.M.E., vol. 33 (1911), p. 245.

actually show the same thing as the Michigan experiments. Mr. De Leeuw drew a false conclusion simply because he compared cutters on the same feed-per-revolution basis. That is, the coarse-tooth cutter appears more efficient merely because for a given feed per revolution it must necessarily take a thicker chip.

The fourth reason follows from a study of the results given by the authors of the paper under discussion. If the data given in the Figs. 1 and 2 are plotted on a chip-per-tooth basis, the curves representing the performance of the four- and eight-tooth cutters cross, thus showing no consistency in efficiency of either. The writer contends that had the conditions been uniform they would not cross but would combine to form a single curve. It is true that the authors' results with the 20-tooth cutter are distinctly higher than those obtained with the other two cutters. The only apparent explanation is that free cutting did not result; that is, the teeth were so close together that deformation of chip must have occurred.

It might be here stated that tooth spacing is not the sole criterion of freedom from interference in cutting, of one tooth in regard to its neighbor. Tooth shape or clearance space is quite a factor.

The 20-tooth cutter referred to is designed according to the formula recommended in the paper summarizing the Michigan experiments, already referred to. That formula was based on the experiments which had then been conducted, and it will be recalled that all the experiments were limited to a width of one-half inch. It would take at least a hundred times as much experimenting, under actual production conditions, to arrive at a formula which could be legitimately defended, and the one in question was suggested merely as a starting point.

The authors' loss of efficiency with a 20-tooth cutter is surprising. Had they taken small steps, from 8 to 20 teeth, a marked decrease of efficiency would have occurred at some definite stage, viz., where free cutting ceased. The energy lost in the machine might be responsible for the apparent deviation from consistency of the authors' results, on a chip-per-tooth basis.

To refer again to the elusiveness of milling-cutter action, in considering the relative merits of coarse- and fine-tooth cutters the fact may be mentioned that Mr. De Leeuw's paper and the paper by Messrs. Hall and Graves both submit evidence leading substantially to the conclusion that coarse-tooth cutters are not best from an efficiency standpoint. And yet if Mr. De Leeuw's results be analyzed by the methods of Messrs. Hall and Graves the converse conclusion would be arrived at. The writer's own statement is that there is no difference, provided comparisons are made on a chip-per-tooth basis.

Referring to Fig. 7, the authors state that the cutting speed is decreased to give a larger feed per tooth. Why should this be done? It is tantamount to admitting that the limitation of the

machine power is such that the most efficiently shaped chip cannot be selected. But even if the limitation of machine power is a serious factor (which it usually is in really obtaining the maximum cutter output), it is yet beneficial to reduce speed and so prolong the life of the cutter.

The authors' formula for minimum power is not minimum cutter power but power for the combined cutter and machine; that is, it is not the criterion of maximum grinding life.

Carl J. Oxford. After a rather close study of this paper the writer's chief criticism is that the authors have attempted to draw general conclusions from what is unquestionably a special case. Practically all of the experiments have been run with one size of milling cutter and with one type of cut. That only mild steel was used throughout is perhaps the most serious objection to applying generally the results recorded.

Under conditions as they exist in many shops today there can be no hard-and-fast rule for numbers of teeth in milling cutters, because the milling machines used and with them the kinds of cuts encountered must in many instances dictate the most successful designs in this respect. Due to the lack of rigidity in many milling machines, and also to their lack of power, it is sometimes necessary to make cutters with a smaller number of teeth than that theoretically advisable.

The writer is of the opinion that many milling machines not only vibrate entirely too much, due to lack of rigidity, but also that many set-ups possess a definite period of vibration. If on such set-ups cutters are used with a number of teeth which, when operated, will synchronize with this period of vibration, then chatter is produced to such an extent that the efficiency of the whole operation is impaired.

While mild steel is a material which lends itself well to experiments such as those conducted by the authors, nevertheless most of the present heavy production work consists of milling alloy steel. Mild steel can be successfully milled at a peripheral cutter speed in excess of 120 ft. per min. and a feed of 0.006 in. to 0.010 in. per tooth, these figures representing average conditions.

Such steels as heat-treated nickel steel, chrome-nickel, chrome-molybdenum, and silicon-manganese steels can seldom be milled at a peripheral speed higher than 50 ft. per min., although it is entirely practical to use the same chip thickness per tooth as for mild steel. If higher speeds are attempted in alloy steel, the result is invariably a rapid destruction of the cutting edge due to overheating. In this case, then, the productivity of a given size of milling cutter is in direct proportion to the number of teeth in that cutter; thus, if in one revolution at a given peripheral speed we can remove, say, fourteen chips 0.010 in. thick, the table feed of the machine will be 0.140 in. per revolution of the cutter, while an 8-tooth cutter will give only 0.080 in. feed per revolution.

The limiting factors for numbers of teeth are then as follows:

1 Rigidity of the machine and holding devices

2 Depth or width of cut and the ability of the machine to deliver power to cutter

3 Structural strength of the cutter.

In designing cutters for general use the importance of these various factors must be weighed and a number of teeth selected which takes them into account, without sacrificing too much of the potential productivity of the cutters.

The writer's experience in highly developed quantity production has not discovered any condition under which a material change in numbers of teeth from those advocated by Professor Airey and himself would be theoretically advisable. The only reason for making cutters with fewer teeth is that many milling machines now in use do not possess the necessary rigidity to carry a heavier oad. When the depth or width of cut becomes very great, many of the older types of machines will not deliver enough power to the spindle to carry a heavy chip on every tooth.

In actual practice we have, for this reason, been obliged to develop a coarser-tooth series of cutters, but the coarseness of teeth does not begin to approach that formerly used and which the authors of this paper advocate. For instance, we have in our regular series 18 teeth in a 3-in. cutter, while our coarse-tooth series will have 14 teeth. Coarse-tooth cutters as formerly made had 8 teeth for this same diameter. We have never found production conditions which would justify this extreme.

In actual production, quality of finish is relatively unimportant, as any surfaces requiring good finish are almost invariably ground. The entire argument regarding finish is only applicable to slabbing cuts. More than 50 per cent of all production-milling cuts are of the type usually known as a side cut. On a cut of this nature there is practically no difference in quality of finish, the fine-tooth cutter being slightly favored.

Carl G. Barth. In a paper read before the Society some eleven years ago[1] Mr. A. L. De Leeuw demonstrated clearly the advantages of coarser pitches (i.e., fewer teeth) in milling cutters than those in general use at the time.

However, some six years earlier the writer had already arrived at the same conclusion by merely reasoning from the knowledge he had of the great advantage of utilizing the power available behind a lathe or planer tool with a heavy cut at slower speed as against a lighter cut at a correspondingly higher speed, since there cannot be any essential difference between the cutting action of a simple tool and that of each tooth in a milling cutter. On the strength of this reasoning the Link-Belt Engineering Company (now the Link-Belt Company) purchased special milling cutters,

[1] Milling Cutters and Their Efficiency, Trans. A.S.M.E., vol. 33, p. 245.

designed by the writer, with fewer teeth than those in vogue at the time. With these cutters the writer made numerous experiments while pursuing his theoretical studies in comparing the action of each tooth in a cutter with that of a lathe cutting-off tool.

On the strength of this work the writer soon constructed a slide rule which, without taking any account of power consumption, for several years enabled him to make some — at the time — astonishing improvements in milling and gear-cutting operations in a number of shops. Some two years ago he finally constructed milling slide rules, which in theoretical completeness are on a par with the slide rules described in his paper on Slide Rules for the Machine Shop as a Part of the Taylor System of Management.[1]

Referring to the matter of power, the writer would like to see most carefully conducted experiments made to determine, for various materials, and first of all for steel and cast iron, the value of the exponent n in the formula

$$P = \text{constant} \times wf^n \qquad [1]$$

for the pressure on a lathe slicing or cutting-off tool taking a cut w wide with a feed f. For with this exponent, which unfortunately is probably a function of f itself, once determined beyond a doubt, mathematical considerations alone will put us as close to the actual power consumption of milling cutters as is at all practically desirable.

The writer's latest milling slide rule for steel was worked up on the strength of the values he assigned to the constant and the exponent in Formula [1], to cover the experiments made by Taylor and Lewis in the early eighties on plain slicing or cutting-off tools. This formula is $P = 81{,}000\ wf^{7/10}$.

The authors of the paper refer to their formula $P = At^{14/15}$ as having the same exponent for the maximum chip thickness as that given for the feed by Taylor in On the Art of Cutting Metals, as the result of his experiments on lathe tools. As the writer is personally responsible both for the experiments and the formulas given by Taylor, he wishes to emphasize the fact that those experiments were on round-nose roughing tools, the curved outlines of which must have an influence on the pressure required to remove chips that undoubtedly means a different exponent for the feed from that for cutting-off tools, with their counterpart in the teeth of a simple slab or groove milling cutter.

Fred A. Parsons. The conclusions reached by the authors appear to be in practical agreement with the results of the tests which were the basis of the writer's paper on Power Required for Cutting Metal,[2] presented at the Annual Meeting of the Society in December, 1922.

[1] Trans. A.S.M.E., vol. 25, (1904), p. 49. [2] See p. 193 of this volume.

Upon the point of decreased power required as the cutter speed is reduced, thickness of chip and other factors being maintained constant, although such a tendency was noted in some of the writer's tests, others of them seemed to leave this factor out entirely. Since these tests were conducted with the idea of obtaining a complete and simple method of predetermining total power, it was felt that all except pronounced factors should properly be excluded as they would complicate the formulas and slide rule without affording a corresponding gain.

Separating the power into the components of cutter power and feed power, as the authors have done, is probably the only practical way of obtaining some of the information desired by the authors; but it complicates matters considerably and, in the writer's opinion, is unnecessary for the majority of such tests, since the total power apparently follows definite laws, regardless of the differences in the efficiency or method of application of the speed and feed trains, provided suitable precautions are taken.

The writer would take some exception to the use of "feed per tooth" or "maximum chip thickness" as the basis for any comparison of cutting tools, since according to his experience such a comparison becomes inaccurate as soon as it involves different depths of cut or milling cutters of different diameters or of a different type. A comparison on the basis of the "average thickness of chip" (A.T.C.) is equally satisfactory for practically all purposes, and has the added advantage that it eliminates differences in depth of cut, diameter of cutter, and type of tool if correctly arrived at. As an instance, the 6-in. side milling cutters and the 3½ in. spiral mills used by the authors may be expected to produce considerably different results owing to the difference in diameter, and this cannot be avoided except by employing a factor which considers the ratio of cutter diameter in relation to cut depth, which A.T.C. does.

The most important contribution to the subject made by the authors appears to be the additional proof which they present of the superiority of coarse-tooth cutters for general purposes and with this the writer's experience is entirely in accord. It should be noted, however, that this does not hold true for face mills, which may properly be constructed with proportionately more teeth than spiral mills, as different principles and limitations are involved.

R. Poliakoff. The authors deal exclusively with the design of milling cutters and not of the milling machine, and, as they state, their object was to study the effect of certain elements of the cutter on power consumption, tendency to chatter, and stresses set up in the machine. Tests of this nature should yield results of a high degree of accuracy, and to secure this a milling-machine dynamometer should be employed.

The writer has already had occasion to describe the milling-machine dynamometer as designed and developed by him, and adopted by The Cincinnati Milling Machine Company, of Cincinnati, Ohio.[1]

With this dynamometer there is no necessity whatsoever of making a preliminary study of the motor driving the milling machine or of its curve characteristics under different conditions of the tests. It gives directly the values of the three components of the cutting pressure due to milling — the horizontal one (a) acting along the arbor, the vertical one (b) acting on the table, and the horizontal one (c) acting at right angles to the arbor (feeding pressure), and thus furnishes a ready means of determining the stresses set up in the machine — one of the problems the authors had in view. Knowing the values of these three components, the full cutting pressure (in pounds) due to milling is $P = \sqrt{(a^2 + b^2 + c^2)}$, and with P known, the power consumption due to cutting is determined very easily, provided the cutting speed V (ft. per min.) is known.

Even the chatter can be measured directly on the dynamometer, corresponding as it does to the different positions of the hands of the dials. And by going a little further and connecting the gages of the dynamometer with recording devices, automatic records of the phenomena studied can be obtained. Such a recording device has been used by the writer in connection with certain dynamometric twist-drill experiments and measurements.

It is the aim now to establish standards for the tools used in cutting metals and the Society has a special standardization committee on milling machines and milling cutters. It can be readily seen that the milling-machine dynamometer affords the readiest means for establishing such different standards as they are called for by the problems of milling. For instance, with its aid cutters can be standardized, a standardized cutter to be one that when taking a standard metal (which means a metal having a unit metal-cutting resistance) will register certain standard readings on the dynamometer. This means also that the size of the cutter and the shape and number of the teeth shall comply with certain standards.

In determining the cutting qualities of a metal, its hardness is of the utmost importance and is usually measured either by the scleroscope or by the Brinell tester. However, as regards milling operations, hardness figures can also be established by testing the metal under standard conditions with the milling-machine dynamometer. In this way a table of milling hardness figures could be compiled and used as a guide for choosing speeds, feeds, etc.

A few words as to the conclusions presented in the paper. The authors limit the effect of spiral angle by the end thrust. In many cases this can be overcome by making the cutter in two halves —

[1] Trans. A.S.M.E., vol. 43 (1921), p. 598.

one having a right-hand spiral and the other a left-hand. The 10-deg. rake angle which the authors find desirable corresponds more or less to generally accepted practice. However, when considering a cutter having no rake angle, one should not overlook the fact that under actual conditions even such a cutter will work as though it had a rake angle. This is because the actual profile of the surface produced by the cutter is not a part of a circle but a cycloidal curve resulting from the relative motion of the cutter and the work, and the tangent to this cycloid will form an angle less than 90 deg. with the front of the tooth, even if this is a part of the radius of the milling cutter. The rake so produced will increase the larger the ratio of feed to speed becomes, and will be different at different points of the cycloidal path of the cutter tooth. It is obvious also from the above that this rake will be different for a roughing cut with a coarse feed and slow speed from what it is for a finishing cut with a small feed and a higher speed.

All this is mentioned to show that the effect of rake angles ought to be studied in connection with a number of other elements which may influence the same rake angle very differently. The authors do not state whether their study of the rake angle was conducted considering these other influences.

The conclusion reached in regard to coarse-tooth vs. fine-tooth cutters again confirms established practice. The authors found that the margin in favor of the former is more pronounced on wide cuts. As chatter is also affected by the width of the work, sometimes one way, sometimes the other, it would appear that this margin of the coarse-tooth cutter may sometimes have to be sacrificed unless provision can be made to affect the chatter in the same direction.

W. A. Knight. Face milling with the usual practice of feeding the work against the rotation of cutter is the poorest and most inefficient of all methods of cutting metals. The cutting edge of a tool must ride the surface a certain amount every time it takes a cut and the cut is comparatively short. This riding is more destructive to the cutting edge than if it were seated in the metal and doing normal work.

A properly ground lathe or planer tool attacks metal in the most logical and efficient manner, and the closer the principles therein embodied are adhered to in the design and use of milling cutters the more efficient will be the milling. There are and always will be numerous cases, as in slot, form, or narrow surface milling or where finish is a determining factor, where the usual form of milling must be continued, but it should be avoided whenever possible. If spiral face milling for large, flat surface work is not obsolete, it should be.

Another type of milling which is very inefficient is side milling: not that of the rotary-planer type with the comparatively large cutter head and inserted cutters, but that with the ordinary side

mill with face and sides at approximately right angles to each other. Here, again, there is too much drag of the cutting edges over the surface without doing any work. In our shop practice we were in the habit of milling the flat faces of a 2-in. cast brass hexagon nut with a high-speed 28-tooth side mill. A single-tooth fly cutter was substituted for this, with the result that we could mill three times as many per grind of tool and have the nuts "mike" closer to uniform size. No reduction was made in either speed or feed. In this case the single tooth did more and better work than the twenty-eight. This was simply going back to first principles and using a tool shaped to attack the metal in a logical way.

THE AUTHORS. The authors are particularly interested in the discussions contributed by Professor Airey and Mr. Oxford, as a part of this investigation was a direct result of their paper On the Art of Milling, in which they recommended that milling cutters should have as many teeth as possible, consistent with sufficient chip clearance.

The authors agree with Professor Airey that the method of measuring power is of fundamental importance in work of this sort, but apparently there is a radical difference of opinion as to what constitutes a proper method. He states that the procedure used in the Brown & Sharpe tests was rejected at the University of Michigan, while it happens that the Michigan methods and apparatus were considered and discarded by the authors as both inadequate and probably unreliable.

The authors also agree with Professor Airey's statements concerning the difficulties and possible errors in the method he mentions of making use of the electrical input, but they are at a loss to understand why the much more simple and accurate variation of this method, which was used in their tests, was not considered. As noted in the paper, observations of the motor input with the spindle running at the proper speed were made before and after each test cut and this amount was deducted from the wattmeter reading during the cut. In this way the greater proportion of the losses, including most of those which vary with different conditions of operation, were immediately eliminated. The relatively small increase in electrical and friction losses caused by the load were determined by a prony brake, and a large number of tests showed that these losses were always practically the same for any given output, irrespective of variation in the no-load wattmeter reading at different spindle speeds. Professor Airey objects that the losses in a prony-brake test are not the same as when cutting metal, and this is technically correct whenever border-line lubrication exists. Even under this condition, with good lubrication such as was maintained during these tests, the maximum error would always be less than three per cent, and this seemed and still seems to the authors hardly worth bothering about in this particular case.

In the Michigan tests three forms of apparatus were used but

the only one on which any actual milling was done was the "torque and thrust measurer." The chief objections to dynamometers of this kind are lack of the rigidity so essential in machine tools, limitation of the tests to very small capacities, and the difficulties in continuously getting reliable readings. The authors built a dynamometer for use in their tests and the results, some of which are included in the paper, simply reinforced the conclusions derived from the other experiments. Their experience along this line, however, makes them feel that it is never safe to rely absolutely on the data given by such apparatus unless it is frequently calibrated or some supplementary figures such as the electrical input into the motor are taken to guarantee that the results are reliable.

This feeling concerning the unreliability of dynamometers was reinforced by the apparent inconsistencies in the work of Professor Airey and Mr. Oxford. In all, fifty-five tests using the "torque and thrust measurer" were reported in their paper, and of these twenty-six were devoted to the effect of different rake angles and only twenty-three to the effect of varying tooth spacing. In the first set their figures show that much more power was required for cutting machine steel than alloy steel on corresponding cuts, while in the second the alloy steel required about ten per cent more than the machine steel. Neither of these comparative results seems altogether reasonable, but a much more serious discrepancy is found when the machine-steel tests in the rake-angle studies are compared with the corresponding tests in the tooth-spacing studies. Here their figures show that the latter required about fifty per cent more power than the former for similar metal on similar cuts with similar cutters, a variation several times greater than it seems reasonable to expect. Until such apparent inconsistencies are cleared up, the authors feel that the methods used in the Michigan tests cannot be fully accepted as reliable, nor their results used as definite proof of any theory of cutter action.

Professor Airey gives four reasons for his conclusion that fine-tooth spacing is superior to coarse-tooth spacing. The first of these is purely theoretical, being based on certain assumptions concerning cutter action, and the only way to determine that all factors are taken into account is to verify the theory by extensive experiment.

The results of the Michigan experiments, given as the second reason, are not conclusive, due to the small number of milling tests devoted to this subject, and the inconsistencies in these tests noted in an earlier paragraph. The large number of special studies made with chip investigators and other apparatus were very interesting and valuable, but they can hardly take the place of experiments where actual milling is done.

The third reason is based, according to the discussion, on experiments described in a paper by Mr. A. L. DeLeeuw, but, as this paper does not include sufficient data to make the comparisons mentioned, the authors presume that Professor Airey intended to

refer to the Cincinnati Milling Machine Company's Treatise on Milling and Milling Machines, from which the cutter tests analyzed in his original paper were taken. Three cutters similar in every respect except for the number of teeth were used in these tests, and practically identical cuts at three different feeds were taken with each cutter. The results showed that the finest-tooth cutter required an average of 36 per cent more power than the coarsest cutter on corresponding cuts. Professor Airey, however, plotted the results, using "cubic inches per horsepower-minute" as ordinates and "chip volumes" as abscissas, and found that the points for all cutters fell roughly on one curve. Hence he claims "the coarse-tooth cutter appears more efficient merely because it must necessarily take a thicker chip," hence "Mr. DeLeeuw's paper submits evidence leading substantially to the conclusion that coarse-tooth cutters are not best from an efficiency standpoint, hence, as a result of this and the Michigan tests (quoting from the original paper), "we have thus demonstrated a priori that fine spacing is superior to coarse spacing, regardless of whether the cutting speed or the power of the machine be the limiting factor."

The authors agree with Professor Airey that the greater power efficiency of coarse-tooth cutters is largely due to the fact that the chip per tooth is larger, but they find it impossible to follow his reasoning beyond this point. The trouble with his curve interpreting the Cincinnati tests is that different outputs in cubic inches per minute are not differentiated, and as a result the point for the maximum output of the coarse-tooth cutter is over two and a half times as far out on the curve as the point for the corresponding output of the fine-tooth cutter. The latter cut required 16.5 hp. on a machine which was probably designed for and equipped with a 15-hp. motor, and, if the feed were increased sufficiently to give the same size of chips as the coarse-tooth cutter, about thirty-five horsepower would be needed. As this is over twice the capacity of the machine, it can be eliminated as impossible.

In his original paper Professor Airey claimed that the power required is independent of the cutting speed provided the same size of chip per tooth is taken, so he recommended, in cases like the above where the capacity of the machine is reached, that the speed of the fine-tooth cutter be reduced until the proper size of chip — here that of the coarse-tooth cutter — is taken, and then the power efficiency of both cutters would be the same and the fine-tooth cutter would last longer because of the slower cutting speed. Unfortunately tests involving varying cutting speeds and varying outputs were combined indiscriminately in the Michigan milling tests, so only work done with a chip investigator could be offered as proof of the above claim. On the other hand, the authors studied this claim by tests in which the feed and hence the output was kept constant and the chip size varied by changing the cutting speed. Professor Airey criticises this procedure in his discussion, but its reason seems obvious. In every case these

tests showed that the coarser-tooth cutters were somewhat more efficient than the finer ones for the same chip per tooth, indicating that the cutting speed did affect the result. The authors have thus shown by definite experimental study of this claim, which is not covered in Professor Airey's milling tests, that the slowing down of the fine-tooth cutter in the Cincinnati experiments to a surface speed of 24.3 ft. per min. to make the chip per tooth the same size as that taken by the coarse-tooth cutter would not have made the power efficiency of the cutters the same. Furthermore, the authors' study shows that the power required by the milling-machine table at this low cutting speed would have been greatly increased and this constitutes another objection to abnormally slow-speed operation.

Even if Professor Airey's theory is assumed to be correct and the power efficiency of the fine-tooth cutter running at 24.3 ft. per min. to be the same as that of the coarse-tooth cutter at a normal cutting speed, it is impossible to put it into effect because of the limitations in the strength of the machine and the arbor. Assuming that 80 per cent of the 13.24 hp. required for the coarse-tooth cutter was actually delivered to the cutter and that the same amount of power would be required by the fine-tooth cutter running at the slower surface speed, the maximum shearing stress in the $1\frac{1}{4}$-in. arbor would be over 56,000 lb. per sq. in. due to the torsion alone, and making no allowance for the increase in this stress due to bending. The thrust along the table, which would have to be taken care of by the lead screw, would be about 14,500 lb. The first of the above figures is far above the safe limit, while the second is probably greater than that for which the machine was designed. The corresponding figures for the coarse-tooth cutter would be 25,500 lb. per sq. in. and 6600 lb., respectively. It is therefore obvious that even if Professor Airey's theory were correct, it would be impossible to get as high efficiency from the fine-tooth cutter as from the coarse- without redesigning the milling machine and arbor to operate under much greater torques and pressures, and it is equally obvious that the Cincinnati tests become what they are supposed to be, a real argument for the superiority of coarse-tooth cutters.

As a fourth reason for his conclusions regarding tooth spacing, Professor Airey states that he plotted on a chip-thickness basis the data from two series of tests where the authors were comparing output with power efficiency and found that the curves for the four- and the eight-tooth cutters crossed. This is shortly followed by the statement that the authors' paper submits "evidence leading substantially to the conclusion that coarse-tooth cutters are not the best from an efficiency standpoint." Now both of the cutters mentioned have several times the tooth space recommended by Professor Airey. The four-tooth cutter was consistently slightly more efficient that the eight- in the authors' studies of the relation of chip size to power efficiency. The average of the studies re-

plotted by Professor Airey likewise show the four-tooth cutter more efficient than the eight-. The power efficiency on every test in the group falls within five per cent of the figure necessary to give smooth and absolutely consistent curves. Finally, the cutter having the number of teeth recommended by Professor Airey required much more power than the two mentioned. In view of these facts the authors find it hard to understand the statement concerning the inconsistencies in their work which would lead one to the conclusion quoted at the beginning of this paragraph.

Professor Airey explains the low efficiency of the cutter having the tooth spacing recommended in his original paper by stating that chip interference or lack of free cutting must have existed. The authors observed these tests very closely and are sure that this did not occur. Furthermore, the low power efficiency started with the finest chips, and it improved as the size of the chip per tooth increased, which is directly contrary to the probable result if chip interference existed. Finally, even these inefficient fine-tooth cutters required less power at corresponding chip thicknesses than any of those reported in the Michigan studies of tooth spacing when cutting machine steel.

As a result of the above analysis of the four reasons presented in favor of fine-tooth cutters, the authors feel that the case remains to be proved, and they are certain that the sweeping general statements in favor of fine spacing made in the original paper by Professor Airey and Mr. Oxford are not justified by the data in hand.

Before leaving Professor Airey's discussion the authors desire to correct an error in his table for determining the roughness of milled surfaces. The title, "Deviation of Cutter from True Concentricity," should be changed to "Run-out" or the figures in the column divided by two. The former term would normally be taken as the deviation of the geometric center of the cutter from its center of rotation, while the latter is generally applied to the variation in the height of the teeth through one revolution of the cutter as shown on a dial indicator. The authors feel that the suggested term is the better one to use as it includes not only the variations due to the eccentricity of the cutter but also those due to the imperfect grinding of the teeth—whose height may differ by as much as 0.002 in. in ordinary shop practice, particularly in cutters having large numbers of teeth.

Mr. Oxford in his discussion presents an interesting and plausible argument in favor of fine-tooth cutters, but it depends for its validity on three assumptions: First, that for every material there is a maximum size of chip which can be taken by each tooth of a cutter consistent with reasonable life between grindings; second, that this figure can be determined with reasonable accuracy; and third, that a properly designed milling machine would be sufficiently powerful to take such a chip on every tooth of a fine-tooth cutter running at a reasonable cutting speed and would be

sufficiently strong and rigid to eliminate chattering and undue strains under such a load.

The difficulties arising under the first and second assumptions are illustrated by the fact that feeds per tooth much heavier than those given by Mr. Oxford for production work in mild steel are frequently carried by coarse-tooth cutters on moderately wide work. On the other hand, the same feed per tooth requiring the same amount of power per inch of width frequently causes the cutter to become dull much more rapidly on narrow cuts. The reason for this appears to be that the number of teeth in contact with the wider work due to the spiral angle of the cutter eliminates the shock on the teeth as they start to cut, and this impact on the narrow cut seems to have more effect on cutter life than the power used per inch of tooth front. This raises a question as to what extent the "energy transformed into heat at the cutting edge" of the cutter is the only "criterion of maximum grinding life," which the authors understand to be the theory back of the conclusions in On the Art of Milling.

Turning to the strength of the machine, feeds per tooth easily carried by coarse-tooth cutters on moderately wide cuts cannot be carried by fine-tooth cutters unless much more powerful and heavier milling machines are built, as illustrated by the Cincinnati tests analyzed in a previous paragraph. However, whether the cuts be narrow or wide, tendency to chatter rather than the power of the machine is generally the limiting factor with fine-tooth cutters. Possibly each set-up has a natural period, but the authors' experience with fine-tooth cutters working near their capacity has been that the number of such periods is extremely large, as frequently chatter could not be eliminated by any reasonable change in cutting speed or small change in feed. The only solution was to greatly reduce the output with the fine-tooth cutter or substitute a coarse-tooth cutter. Of course, the rigidity necessary for the fine-tooth cutter might have been secured by transferring the work from, say, a 3-hp. to a 15-hp. machine, but this is hardly a feasible answer.

Whenever it becomes necessary to put on more teeth in a cutter either to eliminate hammering or to reduce the wear due to chips of too large size taken at normal cutting speeds, the authors feel, as stated in the paper, that the change must be a matter of judgment until the laws of cutting are more fully determined. The authors in no case advocate putting in as many teeth as possible, and they are interested to find Mr. Oxford developing a coarser series of cutters instead of following the recommendations of his original paper by running the very fine-tooth cutters at a slower cutting speed. Under these conditions the authors do not find their practice so far from his in the special case of narrow cuts on the tougher alloy steels where the limiting conditions mentioned above become particularly aggravated. The authors also feel sure that when Mr. Oxford comes to the problem of wider cuts,

he will soon develop a third series of cutters fully as coarse as those used by the authors on slabbing work.

The authors are interested to note that the other discussions are in general accord with the conclusions of their paper, and therefore are omitting any detailed discussion of the points raised.

No. 1897

POWER REQUIRED FOR CUTTING METAL

By Fred A. Parsons,[1] Milwaukee, Wis.
Member of the Society

This paper gives results of an investigation extending over a period of more than ten years, the purpose of which has been to determine the fundamental laws governing milling, turning, planing and drilling operations on the various metals and alloys used in machine construction. In addition to a very large number of tests made on milling machines constructed by the concern with which the author is connected, those reported by Frederick W. Taylor and Professors Bird and Fairfield in the Society's Transactions have been subjected to analysis, the following variables being studied:

1 *The efficiency of the machine*
2 *The rate of metal removal (cu. in. per min.)*
3 *The average thickness of chip before distortion*
4 *The front rake on the cutting blade*
5 *The material being cut*
6 *The spiral angle or shear on the cutting blade*
7 *The condition of the cutting tool, sharp or dull.*

The author's results are presented in the shape of formulas and tables by means of which the power required to machine metal in any given case may be calculated, and an example of their use is worked out in detail. Such calculations, however, are tedious, and the author has accordingly devised a slide rule for the purpose, a description of which forms an appendix to the paper.

THE VARIABLES which may affect the relative power required at the drive pulley of a machine tool are, according to present data, limited to the following, there being no evidence that speed of cutting affects power economy to any marked degree

[1] Chief Engineer, Kempsmith Milling Machine Company.

Contributed by the Machine Shop Division and presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged.

a The efficiency of the machine
b The rate of metal removal (cu. in. per min.)
c The average thickness of chip before distortion (A.T.C.)
d The front rake on the cutting blade
e The material being cut
f The spiral angle or shear on the cutting blade (trifling)
g The condition of the cutting tool, sharp or dull.

It also seems likely that lubrication or flooding of the cutter may affect the power economy, but at present data on this point are not available.

2 In reality the seven variables mentioned in Par. 1 involve

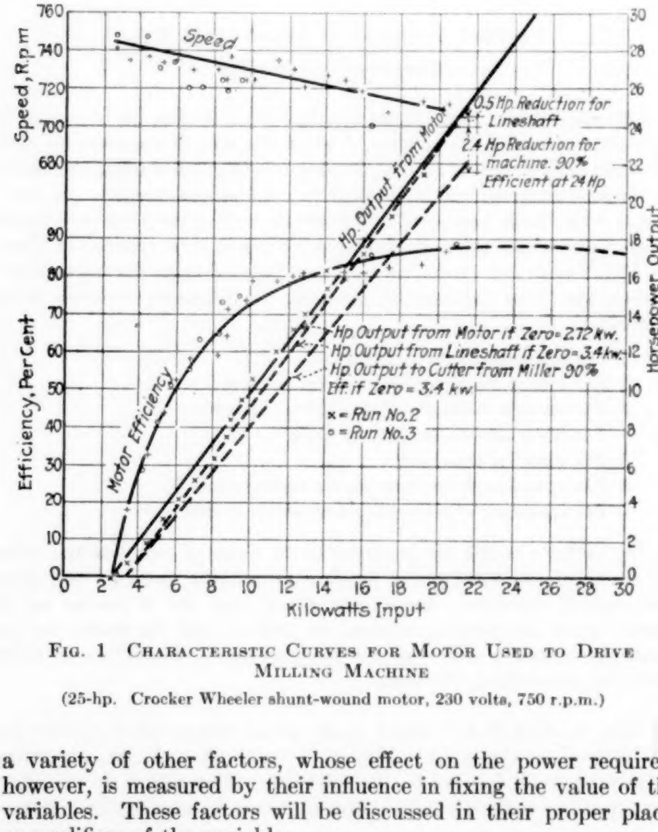

FIG. 1 CHARACTERISTIC CURVES FOR MOTOR USED TO DRIVE MILLING MACHINE

(25-hp. Crocker Wheeler shunt-wound motor, 230 volts, 750 r.p.m.)

a variety of other factors, whose effect on the power required, however, is measured by their influence in fixing the value of the variables. These factors will be discussed in their proper place, as modifiers of the variables.

3 The data available were those of a very large number of tests made at the plant of the Kempsmith Manufacturing Company on various milling machines of their manufacture, and of

other tests on power consumed, notably those of Taylor as given in his On the Art of Cutting Metals, and those of Professors Bird and Fairfield at the Worcester Polytechnic Institute (Trans. Am. Soc. M. E., Vol. 26), these being chosen because the results were listed as "pressure on tool" or as "torque," from which the power to the tool itself could be computed and thus a distinction be made between the power required for cutting and that required for running the machine.

4 To establish an exact curve of motor efficiency for the motor in use a brake test was run (see Fig. 1). Great care was used and the results gave a reasonably smooth curve when plotted. This will be referred to again, however, as it was later found that the curve, though probably accurate, could not be used.

5 Referring to Fig. 1, it will be noted that the points through which the efficiency curve is plotted show decided jumps around 70 per cent efficiency at 7 kw. input. This led to the belief that it might be just as accurate to take the tests involving small power inputs without adding dead load, which is usually done in order to bring the loading to a point where the efficiency curve is not rising rapidly. This was done in the following manner:

6 In Fig. 1 the line "Hp. Output" is a straight line, making it possible to reduce the kw. input directly to hp. output by means of a simple formula, neglecting entirely the value of the motor efficiency except at one point, preferably a point of high input. Such a formula may provide for a shifting zero point, as follows:

$$\{H \div (K - z)\} \times (k - z) = \text{Hp. output}$$

where H = horsepower output at high point
K = kilowatts output at high point
z = measured zero load, kw.
k = kw. reading at any point

7 Now if a small amount, say, 0.5 hp., is assumed as required to run the lineshaft, and a high point, say, 24.5 hp., is chosen on the motor hp.-output line (see Fig. 1), reducing the motor output 0.5 hp. at that point will give a new point at 24 hp. which may be used as the upper end of a line representing hp. output from the lineshaft to the machine pulley.

8 The lower or zero point being determined by a reading taken before each test of the motor, lineshaft, and machine pulley idle, the formula of Par. 6 may then be changed to give hp. output to machine as follows:

$$\left(\frac{24 \text{ hp.}}{21 \text{ kw.} - z}\right)(k - z) = \left\{\begin{matrix}\text{Hp. delivered to} \\ \text{machine pulley}\end{matrix}\right\}$$

9 While absolute accuracy cannot be expected of practical tests involving tachometers and electrical apparatus, results appear to show that this is the most convenient method of determining the horsepower delivered to the machine pulley corresponding to a given kilowatt input to the motor, where the load is small, and

it has been used throughout in these tests. For small net loads the fixed efficiency curve leads to absurd results whenever the zero point varies from the point employed in establishing the curve.

Effect of Machine Efficiency

10 The efficiency of a machine is a variable quantity at different loadings. The power required to run the machine idle and the maximum efficiency of the mechanism at full load may be con-

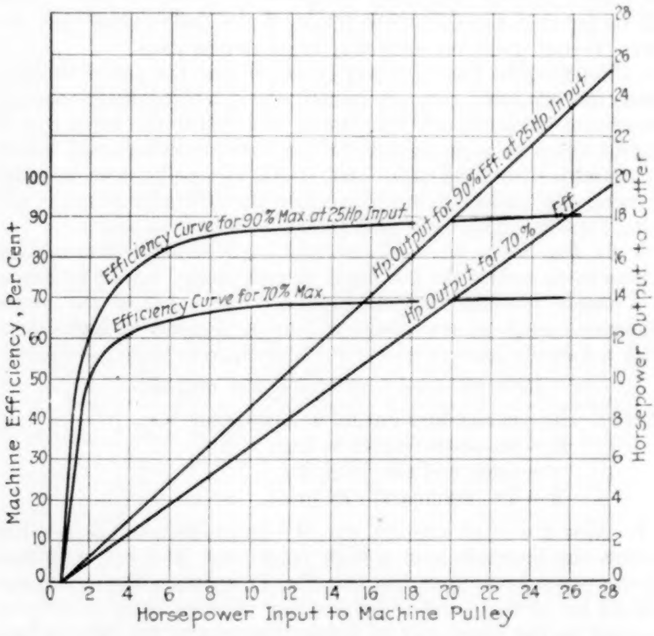

FIG. 2 IDEAL MACHINE. HORSEPOWER OUTPUT AND INPUT AND EFFICIENCY AT VARIOUS LOADINGS

sidered as determinants. If a milling machine, for instance, is set up for a certain cut requiring, say, 1 hp. delivered to the pulley, and the width of the cut should then be increased fifteen times, the horsepower required at the drive pulley will not be increased in the same proportion. This is due to the increased efficiency of the machine at the new and larger load.

11 For consistent results, variations due to machine efficiency must be eliminated. Fig. 2 shows the efficiency at various loadings for two machines, of 90 per cent and 70 per cent maximum efficiency respectively. This chart was plotted as follows:

a The zero point in the efficiency curve was taken as 0.6 hp. from the average results of a number of tests of the power required to run the No. 4 Kempsmith Maximiller with all gearing running but idle

b The high points in the hp.-output lines were determined by a series of tests, which showed that in a milling machine of good design with anti-friction bearings a maximum efficiency of at least 90 per cent may be expected, while in a machine with plain bearings throughout it may be as low as 70 per cent

c Taking these efficiencies as the extremes for milling machines of the two types gave the two lines for hp. output. Readings of these lines at intermediate points gave the values for plotting the efficiency curves.

12 Referring to the first case mentioned in Par. 10, the 1 hp. delivered to the pulley would represent, according to Fig. 2, about $\frac{1}{3}$ hp. output from the machine, that is, to the cutter, on the 90 per cent machine. It is this value which must be increased fifteen

TABLE 1 CONSTANTS FOR USE IN FORMULA OF PAR. 15

Measured kilowatts at zero load	$\frac{21.6 \text{ hp.}}{21 \text{ kw.} - \text{measured zero}}$	Measured kilowatts at zero load	$\frac{21.6 \text{ hp.}}{21 \text{ kw.} - \text{measured zero}}$
2.2	1.15	3.8	1.26
2.4	1.16	4.0	1.27
2.6	1.17	4.2	1.28
2.8	1.18	4.4	1.30
3.0	1.20	4.6	1.31
3.2	1.21	4.8	1.33
3.4	1.23	5.0	1.35
3.6	1.24	...	....

times or to 5 hp. Again, reading from Fig. 2, this would require an input of 6 hp., or six instead of fifteen times the power for the first cut.

13 In these tests corrections for horsepower and machine efficiency were applied by reducing all horsepower values to a value of "hp. delivered to the cutter" or tool hp. This can then be used to obtain a value of "cu. in. of metal removed per hp. delivered to cutter," from which curves may be plotted for varying conditions.

14 There are two ways of obtaining the value of hp. delivered to cutter, as follows:

a (1) Compute the hp. delivered to the machine, as in Pars. 6 to 8; (2) Reduce this according to a chart similar to Fig. 2 in which the lines are determined to suit the machine on which the tests were run

b Establish a line of hp. delivered from the milling machine on the chart (Fig.1) for motor output, so that the hp. output from the milling machine can be read direct from the kw. input to the motor, and construct new formula similar to those in Pars. 7 and 8.

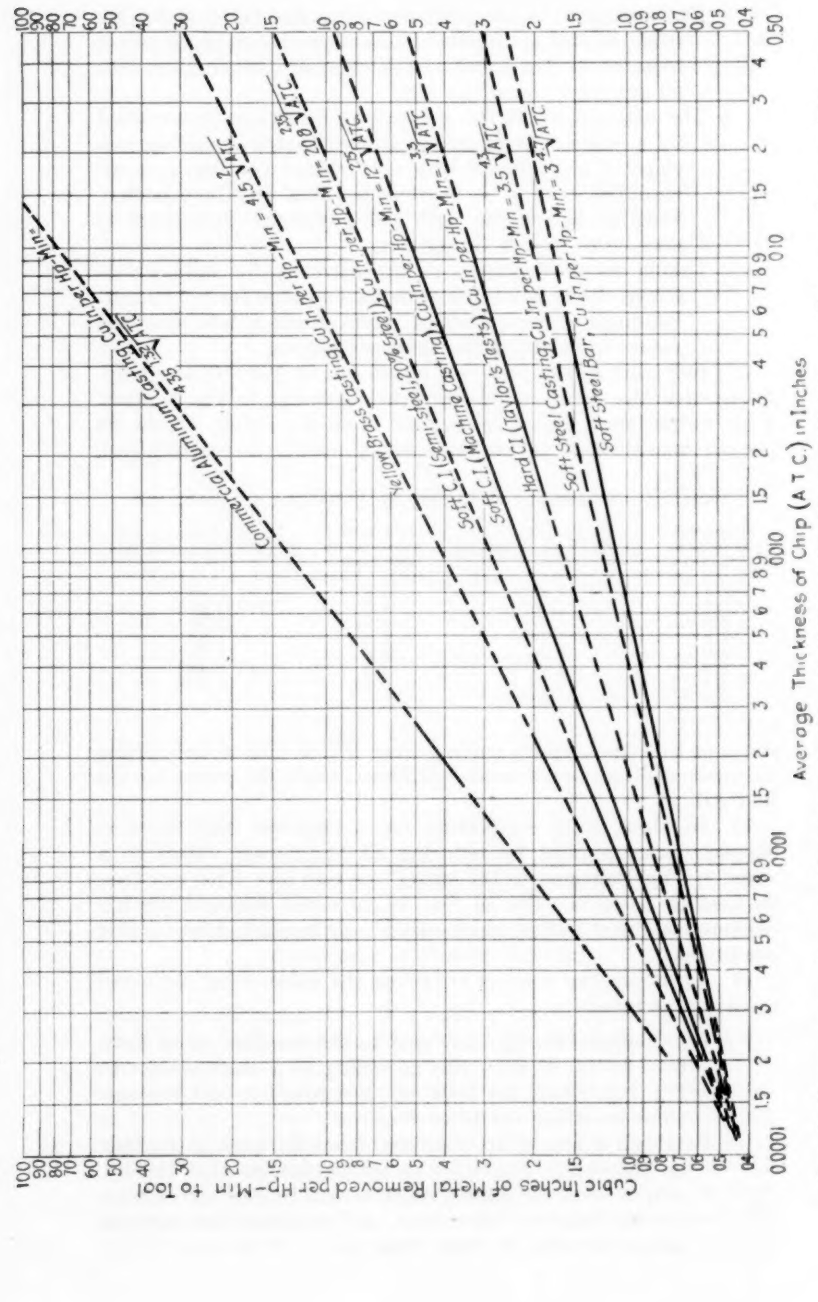

FIG. 3 RELATION OF AVERAGE THICKNESS OF CHIP (A.T.C.) TO CU. IN. PER HP-MIN. TO TOOL FOR VARIOUS MATERIALS
(See note at foot of opposite page)

15 The latter method was the one which was followed. Table 1 gives a list of constants corresponding to the value of the fractional expression in the formula:

$$\left(\frac{21.6 \text{ hp.}}{21 \text{ kw.} - z}\right)(k - z) = \left\{\begin{array}{l}\text{Hp. output from} \\ \text{machine to cutter}\end{array}\right\}$$

This formula corresponds to the line "Hp. to Cutter" in Fig. 1, except that it provides for a shifting of the zero point. Other combinations of machine and motor would of course require other constants.

16 It is recognized that this method might not be so well adapted for machines in which various spindle speeds are obtained by widely different gear trains, but in the Kempsmith Maximillers

TABLE 2 MILLING TESTS OF VARIOUS MATERIALS ON KEMPSMITH NO. 4 MAXIMILLER

Cutter: diam., $4\frac{1}{4}$ in.; no. of teeth, 14; deg. spiral, 25; deg. rake, 10; speed, 48 r.p.m. Cut: depth, 0.250 in.; width, $5\frac{3}{4}$ in.; feed per min., 6.27 in.; feed per rev., 0.13 in.; feed per tooth, 0.0093 in.; A.T.C., 0.0021 in.; cu. in. metal removed per min, 9.

Material	Test Sheet No.	Hp. to cutter	Cu. in. per hp. to cutter	Ratio cu. in. per hp.
Cast iron (soft machine casting) through scale	4005-1	5.71	1.58	1.07
Do., no scale	4005-4	6.06	1.48	1.00
Cast iron (semi-steel 20 %) through scale	4005-7	4.70	1.92	1.30
Do., no scale	4005-9	5.07	1.78	1.20
Steel casting, clean — no scale	4005-11	8.90	1.01	0.68
Steel bar (soft machine steel)	4005-12	10.30	0.88	0.60
Brass casting (yellow brass)	4005-14	4.00	2.26	1.53
Aluminum casting (commercial alloy)	4005-16	1.80	5.00	3.40

the gear contacts are limited to three only for all speeds, and the efficiency for the same loading may therefore be expected to be fairly constant, whatever speed is used.

Effect of Rate of Metal Removal

17 The horsepower varies directly as the cubic inches of metal removed per minute, if all other factors remain constant. This obvious fact has been much obscured by the variations due to other factors, especially variation in machine efficiency at various loadings and variation in proportionate chip pressure for different chip thicknesses.

Effect of Varying the Average Thickness of Chip (A.T.C.)

18 An earlier investigation showed that the power required for milling was closely related to the feed per tooth per revolution and to the depth of cut, the items being considered separately. Later these items were combined into an expression "Average

Note to Fig. 3: — The full lines show range covered by tests. The lines and formulas are for milling cutters with about 10 deg. rake and half dulled by use; for lathe tools with about 15 deg. rake and half dull; and for drills of standard rake and half dull. Allow for any other rake angles and condition of tool when using chart.

Thickness of Chip" which influenced spiral and face mills alike (and also, lathe tools, drills, etc.), gave uniform results over an extreme range of tests, and made proper allowance for the cutter diameter.

19 The A.T.C. affects the power required for removal of metal directly according to a root of its value, the index of the root varying with different materials. Fig. 3 gives the results of a large number of tests and shows the relation for a variety of materials.

20 The tests charted include a considerable number for which the data were taken from the lathe tests run by Frederick W. Taylor and published in his book On the Art of Cutting Metals, as well as some drill tests run at the Worcester Polytechnic Institute. The fact that these tests fall into line with the milling-

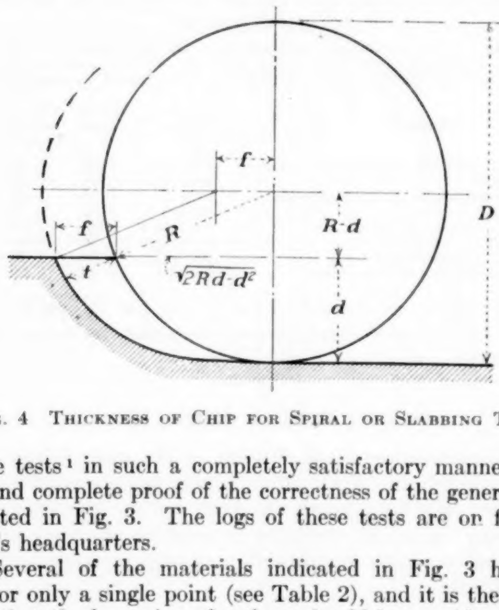

FIG. 4 THICKNESS OF CHIP FOR SPIRAL OR SLABBING TOOLS

machine tests[1] in such a completely satisfactory manner is additional and complete proof of the correctness of the general laws as formulated in Fig. 3. The logs of these tests are on file at the Society's headquarters.

21 Several of the materials indicated in Fig. 3 have been tested for only a single point (see Table 2), and it is therefore not certain that the laws given for these should be considered as fully determined. Since the lines for two materials, cast iron (machine castings) and soft steel, both come to a common point at about 0.0001 A.T.C., it is believed that this may safely be assumed for all other materials also. A single test then locates another, the two fixing the direction of the line for the material tested, and

[1] The complete paper contained four charts showing the relation of A. T. C. to power required for (1) milling and lathe tests on soft cast iron, (2) lathe tests on hard cast iron, (3) milling and lathe tests on soft steel, and (4) drilling tests on soft cast iron. These charts are on file at the Society's headquarters.

thus determining the formula. The lines for aluminum and brass were thus determined. Those for steel and for machine cast iron were determined for nearly the full extent of the chart, and those for the other metals over a limited range, but one sufficient to show that they apparently follow out the theory of a common origin for the lines of all materials.

22 The A.T.C. for the various types of cutting tools must be determined in various ways according to the types, which, for this purpose at least, fall into five classes as follows:

a Spiral or slabbing mills (see Pars. 23–25)
b Formed milling cutters, as gear cutters, etc. (see Pars. 26–27)
c Face mills (1) with square corners on blades or (2) with corners rounded or chamfered (see Pars. 28–37)
d Lathe and planer tools (see Pars. 38–39)
e Drills, counterbores, etc. (see Pars. 40–41)

23 In general, for all milling cutters except formed cutters and face mills it can be shown that the A.T.C. depends upon the feed per tooth per revolution, the cutter diameter, and the depth of cut, the relation being expressed with sufficient accuracy for all practical purposes by a formula derived as follows:

Let D = cutter diameter
R = cutter radius
d = depth of cut
f = feed per tooth per revolution of cutter
= feed per rev. ÷ number of teeth in cutter
A.T.C. = average thickness of chip
t = maximum thickness of chip

Then, referring to Fig. 4, $t = 2f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$

whence A.T.C. $= \frac{t}{2} = f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$

24 The preceding formula is based on the following analysis: In Fig. 4, in the triangle with hypotenuse R the height is $R - d$ and the base is $\sqrt{2Rd - d^2}$. As the triangle with hypotenuse f is similar to the triangle with hypotenuse R,

$$f : t = R : \sqrt{2Rd - d^2}$$

whence $$tR = f\sqrt{2Rd - d^2}$$

but $$R = \tfrac{1}{2}D$$

$$\therefore t = \frac{2f}{D}\sqrt{Dd - d^2} = 2f\sqrt{\frac{Dd}{D^2} - \frac{d^2}{D^2}} = 2f\sqrt{\frac{d}{D} - \frac{d^2}{D^2}}$$

$$= 2f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$$

Since the chip is a regular geometrical figure of maximum height t and ends of zero, its average height (average thickness) is $\frac{1}{2}t$, or

$$\text{A.T.C.} = f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

25 Formula [1] may be still further simplified by tabulating values of the expression under the radical for various values of the ratio d/D, as in Table 3, thus reducing to A.T.C. $= f \times$ constant.

TABLE 3 CONSTANTS FOR VARIOUS VALUES OF THE RATIO d/D IN FORMULA [1]. (SEE PAR. 24)

$\frac{d}{D}$	$\sqrt{\frac{d}{D}\left(1-\frac{d}{D}\right)}$	$\frac{d}{D}$	$\sqrt{\frac{d}{D}\left(1-\frac{d}{D}\right)}$
0.001	0.0314	0.040	0.196
0.002	0.0445	0.050	0.218
0.003	0.0556	0.060	0.238
0.004	0.0630	0.070	0.256
0.005	0.0705	0.080	0.271
0.006	0.0770	0.090	0.286
0.007	0.0832	0.100	0.300
0.008	0.0890	0.200	0.400
0.009	0.0945	0.300	0.459
0.010	0.0995	0.400	0.490
0.020	0.140	0.500	0.500
0.030	0.170	.....	.....

26 For formed milling cutters the formula is (see Fig. 5):

$$\text{A.T.C. for form cutters} = f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)} \times \frac{\text{width of cut}}{\text{length around outline}}$$

This will be apparent if it is remembered that it is the *average* thickness of the chip which is required, and obviously if all the

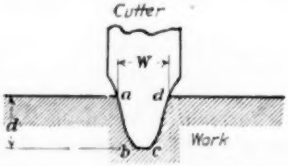

FIG. 5 REDUCTION IN A.T.C. FOR FORM CUTTERS

(W = width of form; d = depth of cut; $abcd$ = length of outline.)

other factors remain the same an increase in the width of the chip as determined by the length of the outline will be accompanied by a corresponding decrease in the average thickness.

27 A reasonable approximation may be had by using a constant representing the approximate value of $W \div$ (length of outline) for various values of d/W, as in Table 4. In practice the A.T.C. for form cutters should first be obtained as if for a spiral mill cut, and then reduced as per formula in Par. 26 or by multiplying by the proper constant from Table 4.

28 For face mills, if the corners of the blades are square when cutting in the usual way, that is, with the work feeding centrally against the edge of the circle generated by the revolving blades (see Fig. 6), it can be shown that the A.T.C. varies according to

TABLE 4 CONSTANTS FOR A.T.C FOR FORM CUTTERS

(Multiply A.T.C. obtained as for a spiral mill by the value corresponding to the ratio of d/W to obtain true A.T.C. See Par. 27)

Value of ratio d/W (See Fig. 5)	0.5	1	2	3	4
Value of $W \div$ approx. length of outline (See Fig. 5)	0.67	0.45	0.24	0.16	0.12

the ratio of the width of cut to the cutter diameter (W/D) and according to f, the feed per tooth per revolution. The A.T.C. for this type of mill is a definite ratio of W/D. (See Table 5.)

29 The constants in Table 5 are determined as follows: A face

TABLE 5 CONSTANTS FOR DETERMINING A.T.C.—FOR USE ONLY FOR FACE MILLS HAVING BLADES WITH SQUARE CORNERS. (SEE PAR. 28)

Ratio W/D (See Fig. 6)	A.T.C.	Ratio W/D (See Fig. 6)	A.T.C.
0.1	$0.999f$	0.6	$0.900f$
0.2	$0.990f$	0.7	$0.853f$
0.3	$0.976f$	0.8	$0.800f$
0.4	$0.959f$	0.9	$0.716f$
0.5	$0.932f$	1.0	$0.500f$

mill cutting a width W' (see Fig. 6) equal to the diameter D would have an A.T.C. equal to half the feed per tooth per revolution; for if the chip mop is a regular geometrical outline whose greatest thickness op is the feed per tooth per revolution (f), and which tapers

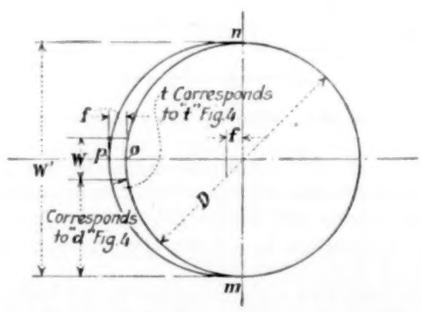

FIG. 6 DIAGRAM REPRESENTING A FACE-MILL CUT

(f = feed per tooth per rev.; D = diam. of face mill; W, W' = various widths of cut; t = approx. end thickness of chip for cut of width W.)

regularly to zero at the ends, the average thickness of such a chip will be $(f + 0)/2 = f/2$. As the width of cut becomes smaller in proportion to diameter D (width W, for instance), the A.T.C. approaches the value of f, until for a zero width of cut the ends of

the chip have the same thickness as the center and the A.T.C. then would be $(f+f)/2 = f$.

30 To determine the exact A.T.C. for intermediate widths of cuts we may use the formula as determined for spiral mills, (Par. 24) and find the value of t (Figs. 4 and 6.) Since t for a face mill is the thickness of the end of the chip, the A.T.C. for any value of W/D will be $(f+t)/2$.

31 The formula for t as determined for spiral mills (Par. 23) holds for t in Fig. 6 if d be considered as the distance from the edge of the cutter circle to the work.

32 For a face mill with any width of cut W, the value of d in the above formula for t may be expressed in terms of D and W by

$$d = \frac{D}{2} - \frac{W}{2} = \frac{D-W}{2} = 0.5(D-W)$$

Then for face mills:

$$t = 2f\sqrt{\frac{0.5(D-W)}{D}\left(1 - \frac{0.5(D-W)}{D}\right)}$$

33 It may be shown that the value of the term $0.5(D-W)$, and therefore the value of the preceding formula and also of the

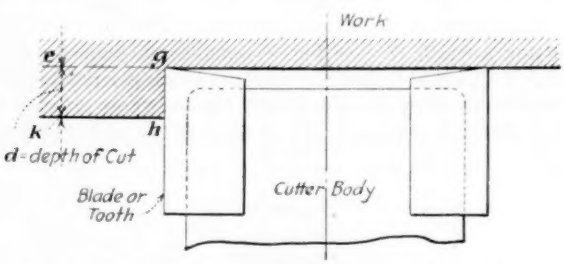

FIG. 7 FACE MILL WITH SQUARE CORNERS ON BLADES

formula A.T.C. $= (f+t)/2$, will be constant for any given ratio of W/D if expressed in terms of f. (See list of such values in Table 5.) The simplified formula for obtaining A.T.C. for face mills having square corners is therefore as follows, the constant being obtained from Table 5: A.T.C. = constant $\times f$.

34 For face mills with a corner radius on the blades, the A.T.C. as determined above for square-cornered mills must be decreased according to constants depending upon the ratio of the depth of cut d to the corner radius on blades r. (See constants in Table 6.)

35 The method of arriving at these constants is as follows: If the corners of the blades are perfectly square (Fig. 7), then the width of chip ek equals the depth of the cut d, and the A.T.C. at all cross-sections of the cut will be the same. If the corners are rounded (Fig. 8), as is almost invariably the case in practice, the

A.T.C. will vary from a maximum for sections of cut not coming on the radius of the cutter blade, as at A, to zero at B. The real A.T.C. for all the sections therefore depends upon the ratio of the

TABLE 6 CONSTANTS FOR USE IN DETERMINING A.T.C. FOR FACE MILLS WITH ROUNDED CORNERS. (SEE PAR. 34)

(Multiply A.T.C. for equivalent cut with face mill having square corners by constant corresponding to value of depth of cut divided by corner radius on blades.)

Ratio $\frac{d}{r}$ (See Fig. 8)	Length of Chip $= A'B$ (See Fig. 8)	Average Area of Chip $= \text{A.T.C.}^S \times d$	Constant $= \frac{\text{area}}{\text{length}} = \frac{\text{A.T.C.}^S \times d}{A'B}$
0.1	0.445r	A.T.C.^{s} × 0.1r	0.225
0.2	0.645r	A.T.C.^{s} × 0.2r	0.31
0.4	0.925r	A.T.C.^{s} × 0.4r	0.43
0.6	1.16r	etc.	0.52
0.8	1.37r		0.58
1.0	1.57r		0.64
2	2.57r		0.78
3	3.57r		0.84
4	4.57r		0.87
5	5.57r		0.90

depth of cut d to the corner radius r, or d/r. This real A.T.C. for round-cornered cutters may be called A.T.C.^{R} and for sharp-cornered ones A.T.C.^{S}

36 Now if the average area of chip is computed by multiplying the average thickness (A.T.C.^{S}) as obtained for face mills with

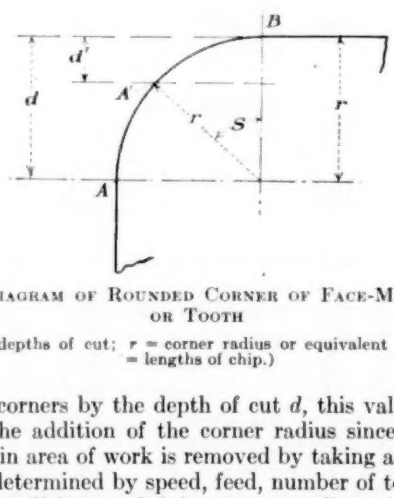

FIG. 8 DIAGRAM OF ROUNDED CORNER OF FACE-MILL BLADE OR TOOTH

(d, d' = various depths of cut; r = corner radius or equivalent bevel; A B, $A'B$ = lengths of chip.)

square blade corners by the depth of cut d, this value will not be changed by the addition of the corner radius since it is obvious that if a certain area of work is removed by taking a definite number of chips, determined by speed, feed, number of teeth in cutter, etc., each chip will have a definite area regardless of its form, and

$$\text{Average area} = \text{A.T.C.}^S \times d$$

The addition of a corner radius, however, will change the length from gh in Fig. 6 to AB in Fig. 8, and the real average thickness or A.T.C.^{R} will therefore change accordingly, or:

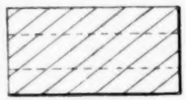

TABLE 7 EFFECT OF FRONT RAKE OF BLADES ON POWER REQUIRED FOR CUTTING METAL

Material cut, semi-steel; depth of cut, ¼ in.; width of cut, 5¾ in.; machine used, No. 4 Maximiller

Front rake B, deg.	Test Sheet No.	Cutter: Type and diam., in.	Cutter: No. of teeth	Cutter: Spiral deg.	Speed r.p.m.	Cut: Feed per min., in.	Cut: Feed per rev., in.	Cut: Feed per tooth, in.	Cu. in. metal removed per min.	Net hp. to cutter	Cu. in. per hp-min.	A.T.C. (Average thickness of chip) in.	Cu. in. per hp-min. revised for A.T.C. = 0.002 in.	Ratio of cu. in. per hp-min.	Ratio $\frac{1}{\text{covers } B}$
0	4004–11	Sp., 3½	6	26	78	4.35	0.056	0.0094	6.25	4.50	1.39	0.0025	130 } 1.35	1.00	1.00
0	4004–19	Sp., 6	10	25	35	3.55	0.103	0.0103	5.1	3.60	1.42	0.0021	140 }		
5	....	......	..	..	..	....	.....	......	.....	....	....	......	....	....	1.10
7½	4004–15	Sp., 4½	11	26	48	6.20	0.131	0.0119	8.9	5.40	1.65	0.0028	1.48	1.10	1.15
10	4004–9	Sp., 4¼	14	25	63	8.65	0.137	0.0098	12.4	7.60	1.64	0.0022	1.60	1.19	1.21
12	4004–13	Sp., 4½	11	26	63	7.35	0.117	0.0106	10.4	6.30	1.65	0.0026	1.59	1.19	1.25
15	....	......	..	..	..	....	.....	......	.....	....	....	......	....	....	1.35
20	....	......	..	..	..	....	.....	......	.....	....	....	......	....	....	1.52
25	....	......	..	..	..	....	.....	......	.....	....	....	......	....	....	1.73
30	....	......	..	..	..	....	.....	......	.....	....	....	......	....	....	2.00

$$\text{A.T.C.}^R = \frac{\text{average area}}{\text{length of chip}} = \frac{\text{A.T.C.}^S \times d}{\text{length}}$$

37 The length $A'B$ in Fig. 8 may be found by the formula —

$$\text{Length } A'B = 2\pi r \times \text{angle } S/360°$$

For any given value of d/r the length expressed in terms of r will be constant, and on this basis for any given value of d/r the value of A.T.C.^{R} will also be constant as expressed in terms of A.T.C.^{S} Evaluating these ratios gives the constants in Table 6, by which the A.T.C. as determined for a given cut with a face mill with square corners must be multiplied to get the true A.T.C. for an equivalent cut with rounded or beveled corners.

A.T.C. for Lathe and Planer Tools

38 After having followed through the above it will be apparent that for lathe and planer tools the A.T.C. will also be dependent upon the form of the tool —

a If the tool has a sharp corner and acts square with the direction of feed, then the A.T.C. = feed per revolution or per stroke

b If the tool is rounded, or stands at an angle, the A.T.C. as determined for the square-cornered tool must be multiplied by the depth of cut and divided by length of the chip, on the same principle as used in determining the true A.T.C. for face mills having rounded corners.

39 There are no accepted standards for tool forms in general use and therefore no table of constants can be given here, though these could be worked out for any standardized set of forms. For the lathe tests of Frederick W. Taylor which are incorporated in Fig. 3, the A.T.C. was modified as above according to the outlines of his standard tools as given in On the Art of Cutting Metals.

A.T.C. for Drills, Counterbores, etc.

40 For counterbores, where the cutting edge is square with the feed, the A.T.C. = feed per revolution ÷ number of teeth.

41 For drills the angle α which the cutting edge makes with the surface of the work must be considered, and the A.T.C. as determined above should be multiplied by the cos α. For a drill ground to the standard angle of 59 deg., $\alpha = 31$ deg. and cos $\alpha = 0.857$.

Effect of the Front Rake of Cutting Blade on Power Consumed

42 Tests are available only for milling cutters, and only up to 12 deg. of rake (see Table 7). The results of these tests seem to show that for milling cutters the power required varies directly as the ratio of the coversed sine of the rake angle, or the cubic inches of metal removed per tool hp. varies inversely in the same way.

43 For milling cutters and lathe tools the front rake may vary considerably, and allowance according to the above should be made when computing metal removed or hp. required from the chart of Fig. 3, which is based upon milling cutters of about 10 deg. front rake and lathe tools of about 15 deg.

44 For drills the rake as determined by the helix angle is not subject to much variation in practice. Fig. 3 being based on drills having the rake angle of the standard twist drill (which is about equivalent to a 15-deg. average cutting rake), no allowance need be made when using it unless the drill to be used is something other than a standard.

Power Required for Various Materials

45 Each material requires a different formula for determining the metal removed per hp-min. or the hp. required for any given

TABLE 8 POWER REQUIRED FOR DULL AND SHARP MILLING CUTTERS

Condition of Cutter	Machine	Test sheet No.	Material	Cutter: Type and diam., in.	Cutter: No. of teeth	Cutter: Spiral, deg.	Cutter: Rake, deg.	Speed, r.p.m.	Cut: Depth, in.	Cut: Width, in.	Cut: Feed per min., in.	Cut: Feed per rev., in.	Cut: Feed per tooth, in.	Cu. in. metal removed per min.	Hp. to cutter	Cu. in. per hp-min. to cutter	A.T.C.	Corrected for	Cu. in. per hp-min. corrected	Ratio sharp to dull
Sharp............	4P. Maxi.	2305–10	C.I.	Face mill 8½ in.	14	10	10	39.5	0.375	6	10.45	0.260	0.019	22.70	7.65	2.96	0.0062	none	2.96	1.45
Dull............	"	2305–11	"		"	"	"	41.5	0.350	5	11.00	0.265	"	22.15	10.90	2.04	0.0059	"	2.04	
Sharp............	"	2323–17	C.I.	Face mill	14	10	10	25.0	0.450	6	7.35	0.294	0.021	19.85	7.25	2.74	0.0069	none	2.74	1.29
Dull............	"	2323–10	"		"	"	"	24.5	0.400	6	7.17	"	"	17.20	8.10	2.12	0.0066	"	2.12	
Sharp............	2P. Maxi.	1217–7	Steel casts	Sp. 3¼ in.	10	25	7	88	0.187	6	4.35	0.050	0.005	4.87	5.80	0.840	0.0012	none	0.840	1.29
Has run —																				
76 in. = 17 min.	"	1217–11	"	"	"	"	"	"	"	"	"	"	"	"	5.95	0.820	"	"	0.820	
132 in. = 30 min.	"	1217–14	"	"	"	"	"	"	"	"	"	"	"	"	6.30	0.775	"	"	0.775	
152 in. = 34 min.	"	1217–15	"	"	"	"	"	"	"	"	"	"	"	"	6.45	0.755	"	"	0.755	
190 in. = 43 min.	"	1217–17	"	"	"	"	"	"	"	"	"	"	"	"	6.75	0.720	"	"	0.720	
228 in. = 52 min.	"	1217–19	"	"	"	"	"	"	"	"	"	"	"	"	6.81	0.713	"	"	0.713	
256 in. = 60 min.	"	1217–21	"	"	"	"	"	"	"	"	"	"	"	"	7.03	0.692	"	"	0.692	
400 in. = 90 min.	"	1218–5	"	"	"	"	"	"	"	"	4.28	"	"	4.77	7.02	0.680	"	"	0.680	
475 in. = 108 min.	"	1218–9	"	"	"	"	"	"	"	"	"	"	"	"	7.10	0.670	"	"	0.670	
650 in. = 147 min.	"	1218–18	"	"	"	"	"	"	"	"	"	"	"	"	7.25	0.660	"	"	0.660	
720 in. = 164 min.	"	1218–21	"	"	"	"	"	"	"	"	"	"	"	"	7.32	0.650	"	"	0.650	
Sharp............	2P. Maxi.	1502–6	Steel bar	Face mill 10⅝ in.	18	6	1	35	0.125	6⅝	10.8	0.310	0.0173	8.95	8.68	1.030	0.0105	none	1.030	1.36
Dull............	"	1502–5	"		16	6½	"	28	0.125	6⅝	11.0	0.394	0.0246	9.10	10.60	0.860	0.0140	A.T.C.	0.760	
Sharp............	2P. Maxi.	1500–17	"	Face mill 10 in.	16	6½	1	40	0.125	6¾	1.44	0.038	0.0024	1.22	1.51	0.810	0.0014	A.T.C.	1.43	2.00
Dull............	"	1500–3	"		"	6½	"	28	0.125	6¾	9.10	0.218	0.0136	7.51	10.70	0.700	0.0080	none	0.70	

cut. See Fig. 3. It is apparently not possible to give any fixed ratio for the power required for different materials as this varies according to the A.T.C., the differences due to varying materials becoming small and negligible when the A.T.C. is reduced to a value of about 0.0001, as previously explained. The tests in Table 2 show the ratios of metal removed when the value of A.T.C. = 0.0021; ratios for other values of A.T.C. can be determined from Fig. 3.

46 Since allowance for various materials has already been made in the chart, it is not necessary to consider this item when computing the hp. required or metal removed per hp. for any given cut, beyond choosing the proper formula or reading from the proper line.

Effect of Spiral or Helix Angle (shear) on the Cutting Blade

47 It is important to distinguish between the action of a spiral angle or shear, as on a spiral milling cutter (which does not make the chip thinner), and the effect of setting a lathe tool on an angle. The latter thins down the chip by increasing its length while its area remains constant according to the feed in use. In the same way the chip thickness is decreased for a form cutter and for a face mill with round blade corners, as previously discussed, or by the angle on the point of a twist drill. The effect of adding a spiral angle to a milling cutter, as far as power efficiency is concerned, seems to be confined to reducing the bumping action of the cut and thereby somewhat reducing the maximum power required. From the standpoint of power required this effect is not important.

48 A test run first with several narrow cutters set up with all the teeth on a line, and again run with the teeth staggered (to give the effect of a spiral cut), showed the power reduced in the ratio of 1.42 to 1.27.

49 The spiral angle is of considerable importance, however, since a large spiral angle will enable fewer teeth to be used in the spiral or slabbing cutter. The only reason for using many cutting teeth in such a cutter is in most cases to reduce the bumping action to a point where the work, jigs, and machine will stand it, and this can be done effectively with fewer cutting teeth if the spiral angle is large. For the same reason a cutter with considerable spiral angle may take a heavier feed than one with small or no angle, though this is often prevented by interference of chips. The increase in the feed rate is, of course, equally as effective as a decrease in the number of teeth in increasing the cutting efficiency.

Effect of Sharp and Dull Cutters

50 The condition of the cutter has a pronounced effect on the amount of power required for removing metal. Tests showing the variations for sharp and dull milling cutters cannot be expected

to show uniform results as there is no standard for dullness. However, a number of tests (see Table 8) show that the power may be expected to increase as much as 40 per cent or more before the appearance of the cut warns the operator that it is time to resharpen in the case of milling cutters.

51 As the economical time between grinds varies with different jobs, it seems best to consider the entire interval from sharp to dull as ten equal units, irrespective of the time involved. The tests — 1217-7, etc., Table 8 — show that the cutter dulls in a fairly regular manner, and on this basis the power required could be considered as increasing regularly to a maximum of about 40 per cent above that required with a newly sharpened cutter, at which point the operator would notice that it needed resharpening from the fact that it was no longer cutting properly.

52 In computing power required or metal removed per tool hp. from Fig. 3, we would, according to the above method, take the results from the chart as being for a half-dulled cutter, i.e., one which had been used for five of the allotted ten units mentioned. The computed power required for any given conditions would then be increased if the cutter had been run *more*, or decreased if the cutter had run *less*, than the time or distance required to half dull it; while if computing metal removed per tool hp. this procedure would of course be reversed, the amount of increase or decrease naturally depending upon the number of units less or more than half dulled up to approximately 20 per cent over and under for the maximum of 5 units.

53 Par. 52 refers only to milling cutters. For drills no data are available, but when allowance is made as above in results of available tests they fall almost exactly on the line corresponding to the half-dulled cutter. Lathe tools, to be run economically, as shown by Taylor in his On the Art of Cutting Metals should be run at speeds which dull them very rapidly as compared with the economical speeds for milling cutters. It is therefore probably safest to consider all lathe tools as *dull* tools, because the time interval between a sharp and a dull tool is, or should be, comparatively very short. Exceptions to this are formed tools and other tools in automatic machinery, etc., where the cost of resetting makes it necessary, as for a milling cutter, to increase greatly the time interval.

54 This completes the consideration of the variables noted in Par. 1 as affecting the power required for cutting metal. From the foregoing may be figured the power required by the cutter for almost any combination of tool and cut. Belt hp. can be computed if the idle hp. and efficiency under a given load are known, though it can be approximated from the last item if the cut represents a fairly large percentage of the machine's capacity.

Example. Find the power required for a gang of 5 gear cutters of 6 pitch, No. 4, $3\frac{7}{16}$ diam., 13 teeth, when cutting in cast iron (semi-steel, 20 per cent steel) with a feed of $7\frac{1}{4}$ in. per min. and at a speed of 56.5 ft. = 63 r.p.m.

a Find A.T.C. for equivalent speed, feed and depth for spiral mill (see Par. 23) as follows:

1 $\text{A.T.C.} = f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$

2 f = feed per tooth per revolution = feed per rev. divided by no. of teeth in cutter = 0.115 ÷ 13 = 0.0088 in.

3 For a 6-pitch gear cutter, depth $d = 0.359$ in., whence $d/D = 0.359/3\frac{7}{16} = 0.104$ in.

4 From Table 3, for $d/D = 0.100$, $\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)} = 0.300$, whence

5 A.T.C. for equivalent spiral mill = 0.0088 × 0.300 = 0.0026 in.

b The value thus obtained must be reduced to obtain true A.T.C. for the form cutter (see Par. 27):

1 True A.T.C. for form cutters = A.T.C. for equivalent spiral-mill cut × constant representing ratio of d to W

2 Width of cut W for a 6-pitch cutter is (measured) about 7/16 in. and the standard depth is 0.359 in., whence

$$d/W = 0.359/0.437 = 0.82 \text{ (approx.)}$$

3 From Table 6, for $d/W = 0.82$, constant = 0.50 approx., whence

4 True A.T.C. = A.T.C. × constant = 0.0026 × 0.50 = 0.0013 in.

c Read from Fig. 3, at the point of intersection of the line of A.T.C. = 0.0013 with the line for cast iron (semi-steel), the cu. in. per tool hp. = 1.25 cu. in. approx. for a cutter with a rake of 10 deg. and the cutter half dulled.

d Make allowance for a rake of 0 deg., as per Table 7 and Par. 42:

$$1.25 \times (1/1.21) = 1.03 \text{ cu. in. per tool hp-min.}$$

e The amount of metal removed per minute by the above cut will be 3.4 cu. in. for the 5 cutters (= feed per min. × area of cut), assuming that the area for a single cutter is one-half the depth times the circular pitch.

f Since we expect 1.03 cu. in. per hp-min., 3.4 cu. in. per min. will require 3.4/1.03 = 3.3 hp.

g The value thus obtained is for cutters half dulled. For sharp or dull cutters this would be decreased or increased 20 per cent, respectively (see Par. 52):

3.3 (1 − 0.20) = 2.65 hp. for newly sharpened cutters
3.3 (1 + 0.20) = 4.12 hp. for dull cutters

h To determine the belt hp. from the tool hp., the constants for idle hp. and for efficiency under cut must be known for the machine on which the cut is to be run; this is merely the reverse of the process of finding the tool hp. (see Pars. 10 to 16). Having once been determined for any given machine, the relation of output to input (see Fig. 2) for the machine can be reduced to a formula.

i On the Kempsmith No. 4 Maximiller, for instance, the 4.12 hp. for dull cutters noted in *g* above would be increased as follows:

$$(4.12 \times 1.08) + 0.6 = 5.05 \text{ belt hp. required}$$

55 While the foregoing method of computing the hp. required for cutting metal may be employed provided a line similar to those in Fig. 3 has been established for the material to be cut, it is much more convenient to use a slide rule in which the data have been incorporated. (See Appendix.)

Summary

56 For milling machines the power economy increases —

a For both slabbing and face-milling cuts:

(1) As the r.p.m. of cutter is decreased, but only if this increases the chip thickness

(2) As the feed per revolution of cutter is increased

(3) As the number of teeth in cutter are decreased (but only if the r.p.m. remains constant)

(4) As the front rake is increased.

b For spiral and slabbing cutters:

(1) As the cutter diameter is decreased

(2) As the depth of cut is increased.

c For face mills:

(1) As the cutter diameter is increased.

(2) As the width of cut is decreased

(3) As the corner radius or chamfer is decreased

(4) As the depth of cut is increased, but this only affects power economy when the blades have a rounded or chamfered corner.

57 For lathes, planers, etc., the power economy increases —

(1) As the feed per turn or per stroke is increased

(2) As the round on tool point is decreased

(3) As the angle of the tool face with direction of feed is decreased

(4) As the cutting rake is increased

(5) As the depth is increased, but this only affects power economy if the end of the tool is rounded.

58 For drills, counterbores, etc., power economy increases —

(1) As the feed per revolution is increased

(2) As the number of flutes or cutting edges is decreased

(3) As the spiral angle or cutting rake is increased

(4) As the drill is ground with a larger included angle of point.

59 While the foregoing points the way to greater power economy, possibilities must in many cases be subordinated to practical considerations. On a miller, for instance, too slow a cutter speed, too few teeth in the cutter and too high a feed, though desirable for cutting efficiency, will cause hammering, and usually the work and jigs will not stand this, even if the machine would do so. Sometimes this can be overcome by using helical mills with large angle of teeth.

60 In certain other details, also, a given set-up may fail in operation even though the computed power is well within the cutting capacity of the machine. Almost any machine may chatter on certain speeds and feeds, even when the cut is fully within the machine's capacity — in fact, often because the cut is too light or

the cutters too sharp to put an initial strain on the supporting structure and take out the slack. More often it is due to synchronized vibrations, which are difficult to avoid for all conditions.

61 Of two spindle speeds, both may be equally efficient in the transmission of power and have equal belt-hp. capacity, yet the gear leverages and bearing and shaft stresses by which one is obtained may be excellent, while for the other they may be very poor, causing unsatisfactory cuts, chatter vibration, and failure.

62 As another instance of practical limitations (though this applies only to spiral mills) it might be supposed that more teeth in the cutter would give equal power economy with greater production per unit of time, provided the feed was increased to give the same average thickness of chip, because more chips would be cut per minute by the greater number of teeth. However, not only is there danger of chip interference, but if it be considered that the r.p.m. for a given cutter is limited for any given material, and again that the feed per revolution is limited in most cases by the finish required — which is generally accepted as being determined for spiral mills by revolution marks and not by tooth marks — it will be seen that a point of r.p.m. and of feed per minute is soon reached where the only way left to increase the average chip thickness and obtain greater economy is to reduce the number of teeth, the only limit in this direction being, as before mentioned, the hammering action of the cutter. As the cut approaches the limit of the machine power capacity the advantage of few teeth in the spiral mill becomes very marked in its effect upon production.

APPENDIX

SLIDE RULE FOR DETERMINING THE POWER REQUIRED FOR CUTTING METAL

63 While the power required for cutting metal may be computed directly from the formulas and charts given in the paper, the slide rule offers a simpler method. The entire process is adaptable to the slide rule, except the determination of belt hp. from the discovered value of tool hp. The arrangement shown in Fig. 9 seems to be the most convenient. The rule was constructed according to the methods described below.

64 The magnitude of the logarithmic scale for the tool hp. was determined according to the desired length of the finished slide rule (in this case about 12 in.) and the range it was desired to cover (in this case 0.5 to 70 tool hp.). This scale was then laid out, using proportional dividers set from an ordinary slide rule.

[NOTE: Where "magnitude" is used in the following, it refers to the spacing of the graduations and not to the full length of the scale, which determines the range covered.]

65 The magnitudes of the other scales were determined from the first one. The length of scale *B* for sharp and dull cutters, for instance, would be the distance from 1 to 1.4 on the hp. scale *A*, the difference in tool hp. for a sharp and dull cutter having been determined as 40 per cent. Scale *C*, for front rake angle on blades, was likewise determined. Scale *D*, for cubic inches of metal removed, is of the same magnitude as scale *A* for tool hp. since tool hp. varies as cubic inches of metal removed per minute.

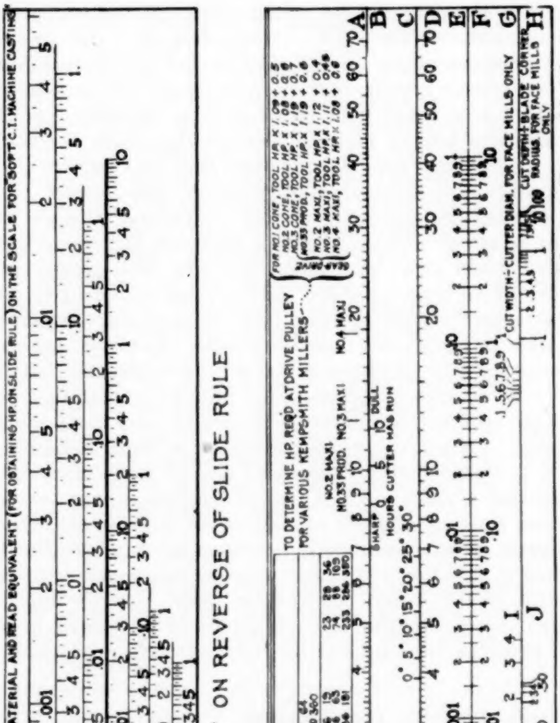

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Fig. 9 Slide Rule for Determining the Power Required for Cutting Metal

(An example of the use of this slide rule is given at the top of the following page.)

EXAMPLE: Find size of miller and belt hp. required for a cut 6 in. wide, ¼ in. deep, 3⅝ in. feed per min. in soft steel, using a slab mill 4¼ in. diam., 15 deg. rake, 14 teeth, at 60 ft. = 54 r.p.m. cutting speed.

(1) Find average thickness of chip as follows: Use scales *I* and *J* for slab mill, set arrow for slab mills (scale *I*) over value of cut depth ÷ cutter diam. = 0.059 on scale *J*. Feed per tooth per revolution = 3⅝ ÷ (14 × 54) = 0.0048; over 0.0048 on scale *F* read 0.00115 on *E* = A.T.C.

(2) Find C.I. equivalent of soft steel A.T.C. 0.00115. On chart on reverse of rule (upper half of Fig. 9) locate 0.00115 on scale for soft steel, and in line with this read 0.0004 on scale for C.I. machine casting.

(3) Determine machine required. Set all scales central, move scale *E* until 0.0004 matches arrow on *F*; over 5.4 cu. in. per min. (= 6 × ¼ × 3⅝) on scale *D* set 15 deg. rake on scale *C*; over 10 for dull cutter on *B* read 7.4 tool hp. on *A*. A dull cutter will therefore require a No. 2 Maximiller or a No. 33 production miller, though certain speeds on No. 3 cone could be used as long as cutter was sharp.

(4) Determine belt hp. required; use constants in upper right-hand corner according to machine. For a No. 2 Maximiller, belt hp. = 7.4 tool hp. × 1.12 + 0.4 = 8.7 hp. for this cut.

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66 Since for each different material the relationship between A.T.C. and tool hp. is expressed by a formula involving a different root of the A.T.C. (see Fig. 3), each material requires a different magnitude for scale *E*. As scales *F*, *G*, *H*, *I*, and *J* must be determined from *E*, it was not convenient to use interchangeable sets of slides, for at least one of these scales would fall on the solid part of the rule.

67 This difficulty was overcome by using a reference scale, pasted on the back of the slide rule. Soft cast iron (machine casting), was selected as a standard for the slides on the rule because it was about midway between the very long scale required for aluminum, and the very short one required for steel.

68 Having decided this point, the magnitude of scale *E* to represent the relation between tool hp. and A.T.C. when cutting soft cast iron (machine casting) was determined from the relative exponent in the formula for this material. For C.I. (machine casting),

$$\text{Cu. In. per Hp-Min. to Tool} = 12\sqrt[2.75]{\text{A.T.C.}}$$

or

$$\text{Cu. In. per Min./Tool Hp.} = 12\sqrt[2.75]{\text{A.T.C.}}$$

from which

$$\text{Tool Hp.} \propto 1/\sqrt[2.75]{\text{A.T.C.}} = 1/(\text{A.T.C.})^{0.363}$$

The fact that the ratio is inverse does not concern the magnitude of the scales, which will be in the ratio of 1 for tool hp. and 1/2.75 or 0.362 for A.T.C., since this is the ratio of their exponents. The scale was laid out with proportional dividers, working from scale *A*.

69 The magnitude of *E* now having been determined, the various reference scales for other materials may be determined in the same way. The formulas for the various materials (Fig. 3) show that the exponent for tool hp. will remain the same for the various formulas, being 1 in all cases; the exponent for A.T.C., however, varies according to the material, as follows:

Aluminum Casting	$\sqrt[1.32]{\text{A.T.C.}} = (\text{A.T.C.})^{0.76}$
Yellow Brass Casting	$\sqrt[2]{\text{A.T.C.}} = (\text{A.T.C.})^{0.5}$
Soft C. I. (Semi-Steel 20 per cent)	$\sqrt[2.35]{\text{A.T.C.}} = (\text{A.T.C.})^{0.42}$
Soft C. I. (Machine Casting)	$\sqrt[2.75]{\text{A.T.C.}} = (\text{A.T.C.})^{0.36}$
Hard C. I.	$\sqrt[3.3]{\text{A.T.C.}} = (\text{A.T.C.})^{0.303}$
Soft Steel Casting	$\sqrt[4.2]{\text{A.T.C.}} = (\text{A.T.C.})^{0.238}$
Soft Steel (20 C)	$\sqrt[4.7]{\text{A.T.C.}} = (\text{A.T.C.})^{0.213}$

70 The various reference scales will vary in magnitude according to the exponents of the A.T.C. The magnitude of scale used on the face of the rule for slide *E* and which corresponds to the correct magnitude for this slide rule for determining the hp. required for C. I. (machine casting) need not be used so long as the relative magnitudes of the various scales are kept in proportion to the exponents given in the preceding paragraph.

71 Scales *A* to *E* would be all that are required for determining hp. to tool after A.T.C. has been found, but as the value of A.T.C. for milling cutters is not readily computed from the formulas, the scales *F*, *G*, *H*, *I*, and *J* are added to the slide rule to facilitate this calculation. Scale *F* for feed per tooth per revolution is of the same magnitude as scale *E*, the exponent of *f* in the various formulas for computing A.T.C. being always the same as that for A.T.C.

72 Scales *G*, *H*, *I*, and *J* are determined in a similar manner to scale *B*, except that the quantity which represents the effect of a given change in the value of the various terms is laid off on scale *E* instead of on scale *A*. There is no scale representing a variation in spiral angle, as the effect is small and apparently can be neglected.

73 The magnitudes of the scales have thus been determined, but not their position. Any scale may be located either upon an upper or lower edge of any slide without affecting the final result, as this merely changes the order in which the variables are considered. The direction in which they read, however, must be reversed. If the direction in which the scale reads is reversed without such a shifting in its location, the effect upon the final reading would be the same as changing from a direct to an inverse relation, in the equivalent formulas.

74 From these two facts it was found possible, by trial, to locate the scales on the slides in such a way that the effect upon tool-hp. scale of shifting the scales *B*, *C*, *D*, and *E* and the effect upon the A.T.C. scale of shifting the scales *F*, *G*, *H*, *I*, and *J*, were the same as that demanded by the relationship of the various terms in the formulas involved and yet have them all read in the normal manner, that is, from left to right.

75 To locate them endwise on the slides in proper relationship to each other and, where necessary, to locate reference arrows such as the one on scale *F* to which scale *E* is set after the A.T.C. has been determined, a problem was worked out by means of the formulas and the corresponding values adjusted on the slides until the result was the same.

76 On the reference scales for various materials on the back of the rule all the logarithmic scales start with a value of 0.0001 on the same line, since all materials seemed to give an equal metal removal per hp. at about 0.0001 A.T.C. If this were not so it would not have been easy to construct one slide rule for various materials, as the reference scales on the back of the rule would necessarily have to be staggered until the points where equal metal removal was obtained were above one another, and instead of a single index arrow for setting scale *E* it would have been necessary to have one for each material having a different value of A.T.C. for the point of equal metal removal per hp. In view of the above facts, however, a single starting index arrow answers for all, so long as it is located opposite the value A.T.C. = 0.0001. For any other location there would have to be an arrow for each material.

77 The fact that the rule will not compute belt hp. does not prevent a point being established representing the maximum tool hp. available for a given machine which will serve to determine whether a given cut may be run on that machine. Cone-driven machines have a different power capacity for each speed, and the best way of representing the maximum capacity seems to be to list the various speeds directly over the point on the hp. scale which represents the maximum for that speed. This has been done on the scale shown herewith for the various Kempsmith cone-driven millers. The above points were established by working backward from the belt hp. figured on the basis of 60 lb. effective pull per inch of width and using the constants in the upper right corner of the rule. These constants were established as described in the notes on the cutting tests.

78 Directions for the use of the rule are fully explained in the example and notes printed on the rule itself. First establish the value of A.T.C. which corresponds to the tool and conditions of cutting for face-mill cuts by setting scales *F*, *G*, *H;* for spiral mills and form cutters, by setting scales *F*, *I*, *J;* and for drills, lathe tools, planer tools, by computing the value according to the data printed upon the rule. Next determine the equivalent of this A.T.C. for the material being cut in terms of soft C. I. (machine casting) from the scale of equivalents upon the back of the rule. The equivalent can be read on the scale for soft C. I. (machine casting) directly above or below the location of the true value on the scale for its own proper material. The tool hp. required for the cut is then determined by locating the equivalent value thus found on scale *E* and setting this value over the arrow. If the scale *C* is then set with the correct rake angle over the correct value of cu. in. per min. on scale *D*, the tool hp. can be read for sharp or dull cutters on scale *A*. When the quantity thus found falls above the point of maximum tool hp. for any given machine, it indicates that a larger machine must be used or the cutting conditions changed to require less power.

DISCUSSION

Benjamin P. Graves and James A. Hall. The author is to be congratulated on his success in bringing together the results of so many and varied experiments into one report, and on finding a basis on which they can be combined into a set of equations governing the art of milling. The tests being given in a form independent of any variables due to the machine or drive enable them to be checked against the work of other investigators. The writers have been making at the Brown & Sharpe Mfg. Co., some investigations of milling along somewhat different lines, but the data were taken in such detail that the tests may be figured along the lines of the author's paper.

The method of deducting the zero kilowatt reading from the reading during the test and multiplying the difference by a constant to give the horsepower delivered to the tool as described by Mr. Parsons, was used by the writers. We differ, however, in the method of determining this constant. In Fig. 1, horsepower output from motor is drawn as a straight line on the basis of 4 hp. to 3 kw. over the zero kilowatt reading, or practically a 100 per cent transfer of all added kilowatts to horsepower. While the losses in field coils, bearings, etc., which make up the no-load input of the motor are practically constant at all loads, the electrical losses in the armature are not, and may amount to a full kilowatt additional at full load in a motor of the size used. We, therefore, question the kilowatt-horsepower line in Fig. 1.

Formulas for determining the factor by which the kilowatt difference is to be multiplied to find the horsepower delivered to the cutter are given in Pars. 6 and 15, being based on the assumption that the horsepower to the cutter is always the same at full-load input to the motor irrespective of the no-load kilowatt reading. This may be represented by a series of kilowatt-horsepower-to-cutter lines starting at different points on the no-load line and

coming to the same point at full load. This gives a different conversion factor for every zero kilowatt reading, as shown in Table 1 and Par. 16. Our investigations at the Brown & Sharpe plant indicated that the no-load kilowatt inputs are required for bearing friction, electrical losses that remain constant as the load increases, etc., and that the kilowatt-horsepower-to-cutter curves should be a series of parallel lines giving the same conversion factor for all no-load kilowatt readings. In our tests we found a conversion factor of 1.25 from motor kilowatt input to spindle horsepower up to one-third of the motor capacity, with a decreasing figure for the higher loads as the electrical losses increased.

We agree with the statements in Pars. 10 to 13 that machine friction losses must be deducted to obtain consistent results in tests. However, we do not understand how the author arrived at a machine efficiency of 90 per cent. The power delivered to a milling machine goes either to the spindle or to the table. The spindle drive may easily have an efficiency of over 90 per cent. The table drive, however, to insure instant cessation of motion when the clutch is thrown out, must be limited to an efficiency of 50 per cent under the most favorable condition of lubrication when operating on a high-speed quick return, and has a much lower figure at normal cutting speeds. This may not be serious when the table takes about five per cent of the power, but it is easy to conceive of a condition where the table might absorb such a proportion of the power as to change considerably the assumed figure of 90 per cent for the whole machine. In the Brown & Sharpe tests, the table and spindle were driven separately, and by different motors. The results given below are based on the power required by the spindle alone.

In Par. 21 it is stated that since the lines for steel and cast iron meet in a point at an average chip thickness of 0.0001 in. and a cubic inch per hp-min. of 0.4, this can be assumed as a common point for all metals. That is, for instance, a 20-tooth cutter, $3\frac{1}{2}$ in. diameter, running at 100 r.p.m., and a depth of cut of 0.025 in. would require the same power at a feed of 2.4 in. per min. on any metal that it might be cutting. If the feed were reduced to 2 in. per minute it would require more power to cut any of the weaker metals than to cut hard steel. This does not seem reasonable, and makes this assumption a point of weakness requiring some defense.

As noted earlier, we have plotted over 100 tests on the basis used by the author. All these tests were made on mild steel and with cutters having ten degrees rake. A line plotted as the average of the tests corresponds with the formula

$$\text{Cu. in. per hp-min.} = 1.53\sqrt[12]{\text{A.T.C.}}$$

If the eight tests at the lower end of Mr. Parsons' diagram are omitted, the Kempsmith tests fit our formula about as well as they do the one assigned to them.

This leaves the Taylor lathe tests practically all above our

line. Taylor interpreted his tests to mean that the vertical pressure on the tool varied as the first power of the depth, and as the 14/15 power of the feed per revolution. For any series at one depth this would make the cubic inches per hp-min. vary as the 15th root of the average chip thickness. We feel that these tests should be represented by a line parallel to and higher than ours rather than by one line that is common for both lathe and milling nuts.

The above criticisms are intended to add to the value of this paper. The discussion of the technique is taken up to draw more attention to this subject so that future methods adopted may appeal to all as essentially accurate. We feel that the great majority of Mr. Parsons' tests are close enough to the actual figures to be accepted without question. While we do not agree with all his conclusions, we certainly feel that great credit is due him for reporting these tests in such a form that they can be combined with similar work of other investigators on other machines.

Carl J. Oxford. The results obtained by the author form an almost complete corollary to the data in the paper [1] prepared by Professor Airey and the writer. Although the author has arranged his results in a different form to facilitate computations of power for commercial purposes, the underlying principles are shown to be the same. It is found that chip thickness is the real fundamental index to the energy required for removing a given volume of metal. Whether thick chips are produced by increasing the linear feed per revolution, or by decreasing the number of cutting edges for a given feed, must depend on practical considerations such as the nature of the cut and the finish desired. In either case, however, metal is removed more efficiently in thick chips than in thin ones. The writer believes that the greatest number of cutting edges compatible with the conditions encountered should always be used to decrease the cutting speed and prolong the life of the tools.

The author's conclusions regarding the lack of power economy due to spiral teeth coincide with the results in the paper mentioned above. This may not hold true, however, for extreme spirals such as those used in helical cutters, although much of the apparent power saving with these cutters is doubtless due to the comparatively small number of teeth and the consequently heavy chip per tooth.

Only in one respect can the practical value of the results obtained by the author be questioned, namely, his failure to consider the saving of power due to the use of lubricants. This is not a criticism, as it is realized that to cover this field an amount of work comparable to that already done would be necessary. It is only to be regretted that no such data are available. It should be noted that the lubricating properties of cutting fluids are en-

[1] On the Art of Milling, Trans. A.S.M.E., vol. 43, p. 549.

tirely distinct from their cooling properties. The function of a cooling fluid is to avoid overheating the cutting edges, while lubrication distinctly decreases the power required for a given cut. That lubrication would seriously affect the calculations of power as outlined by the author is indicated by the statement that we

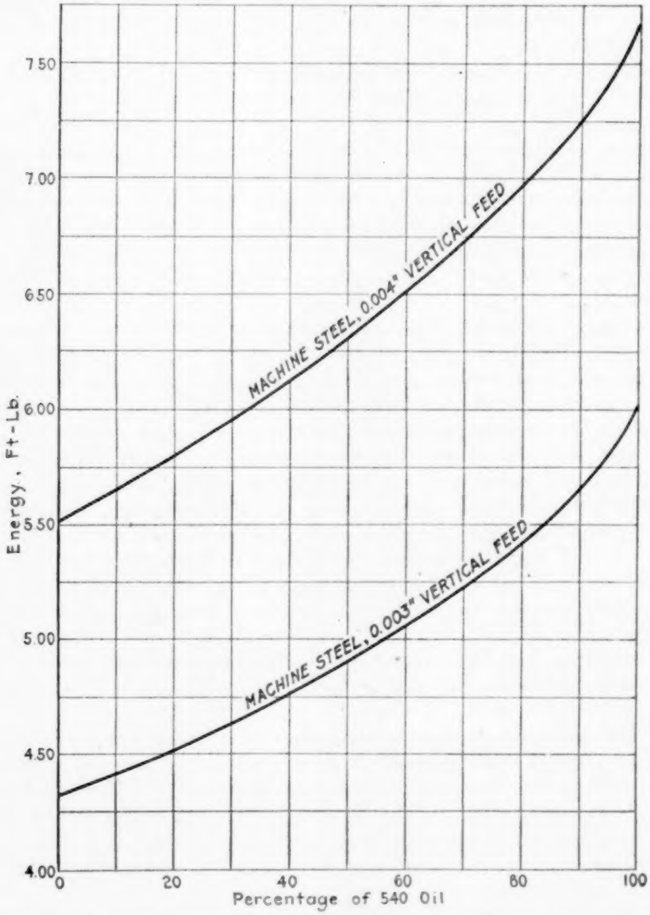

Fig. 10 Energy Required in Milling, Using Different Lubricants

have found extreme variations of over 30 per cent in energy required for a given cut when using different lubricants. The general appearance and viscosity of the two lubricants showing the greatest variation is practically the same. The curve Fig. 10 represents energy in foot-pounds for a given material and with the same chip thickness, but using different lubricants.

While the investigation of lubrication of cutting tools is by no means complete, sufficient data have been accumulated to permit the statement that this factor must be taken into account when computing the power required for metal cutting.

Carl G. Barth. While the experiments described in the paper represent on the whole a more practical method of attack than some of the methods heretofore described, the writer is disappointed that the author did not adopt the same method of eliminating all the electrical and the greater part of the frictional losses in his experimental machinery than the writer did in his own experiments on cutting pressures on lathe tools, as described in Mr. Taylor's paper On the Art of Cutting Metals. Even the author's painstaking efforts to estimate and allow for all frictional resistances in arriving at the net power consumption by the cutting tool itself leave room for doubt as to the reliability of the ultimate results.

The author's experiments deal with power consumption alone. His statement that the experiments have apparently disclosed the fundamental laws governing not only milling, but also turning, planing, and drilling, is curious, for the reason that he makes no mention of feeds, depth of cut, etc. in relation to speeds and economical duration of cutting time in materials of varying degrees of hardness. Such knowledge, as pointed out by Taylor, is of primary importance in practical metal cutting, and a knowledge of power is of only secondary importance. In the author's slide rule the available power alone is represented, in a manner to ascertain whether such power is sufficient for a certain cut, i.e., depth of cut, feed, and speed. The writer has for eighteen years used slide rules with fair success for determining the feed and speed in milling steel and cast iron, on the basis of shape of cut, number of teeth in the cutter, diameter of cutter, and hardness class of the material to be cut. Within the past two years he has added to this a fair knowledge of the power consumption. This knowledge was acquired by a mathematical consideration of the power required by a lathe cutting-off tool.

The author, by the manner in which he uses the writer's results for cast iron, indirectly accepts these as at least as reliable as his own, although they effectively disprove his conclusion that all materials offer about the same resistance to any cutting tool whose effective feed, i.e., chip thickness, is just 0.0001 in. For a certain size of lathe roughing tool the writer obtained a pressure represented, for a certain grade of hard cast iron, by the formula $P = 69{,}000\ D^{\frac{14}{15}}F^{\frac{3}{4}}$, and for a certain grade of soft cast iron by $P = 45{,}000\ D^{\frac{14}{15}}F^{\frac{3}{4}}$, neither grade being the extreme of its kind. For equal depths of cut and feeds the ratio of these expressions is 1.533, and hence also for a feed of 0.0001 in. While he has never dared to assert that these formulas would hold good for a feed so much less than the smallest experimental one there is no reason

to doubt that the ratio they give is approximately correct. For cast irons of extreme hardness this ratio would unquestionably be at least 2, and perhaps 3.

The author also bases his conclusions and final formulas on a false use of the average thickness of the undistorted milling chip, which he assumes to be one-half of its maximum thickness. A milling chip is only of approximately uniformly increasing thickness, and for a ratio of d/D (as used by the author) that approximates $\frac{1}{2}$, as, to be sure, it actually does in certain kinds of end milling, the thickness drops off rapidly from a straight-line increase toward the heavy end, the thickness at any point being practically directly proportional to the sine of the angle between the point of zero thickness and the thickness considered.

Further, figuring the thickness for lesser values of d/D as if the chip were a straight-line wedge, and hence having an average thickness equal to one-half its maximum, this average thickness is not a true measure of the total work done in removing the chip, except for a metal for which the exponent for a uniform chip thickness is unity. We know of no such material. But it can be shown that for a material whose resistance to cutting is proportional to t^n, where t is the chip thickness and n is a fraction, an ideal uniform chip thickness t_i that would require for the removal of the chip the same amount of work as a straight taper chip of equal length and maximum thickness t_m will be

$$t_i = \frac{t_m}{(1+n)^{\frac{1}{n}}}, \text{ whence } \frac{t_i}{t_m} = \frac{1}{(1+n)^{\frac{1}{n}}}$$

Of the following values of t_i/t_m, the smallest one is within the smallest one obtained by Mr. Parsons, viz., for aluminum:

$n =$	1	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$
$t_i/t_m =$	0.5	0.474	0.444	0.41

The assumption as to the average thickness of chip from a lathe roughing tool, which the author has used in arriving at the general conclusion that the power consumption per cubic inch is the same regardless of the kind of cutting tool used, does not seem warranted. Neither does the use he has made of the writer's experiments in cast iron seem fully warranted as a confirmation of his own results. If the assumption held good, the cutting pressure for equal depths of cut and feed would be the same for the fillet tool, Fig. 11, and the straight-edged tool, Fig. 12, with their equally long cutting edges.

The fillet tool has to exert a greater reaction against its chip because this comes off with much more distortion than the chip from the other tool. The ideal average chip thickness of the fillet tool is less than its arithmetical average, as was brought out above for a plain milling chip. Thus it is evident that the power consumption for equal amounts of metal removed must be more for

a fillet tool, and hence also for a round-nose roughing tool, than for a straight-edged tool.

The above arguments hold good in a modified manner for form milling cutters, which are further complicated by the fact that each point on the cutting edge of a form cutter works at a shallower depth of cut and on a smaller diameter than the extreme points, a fact ignored by the author. For such cutters there is a variable

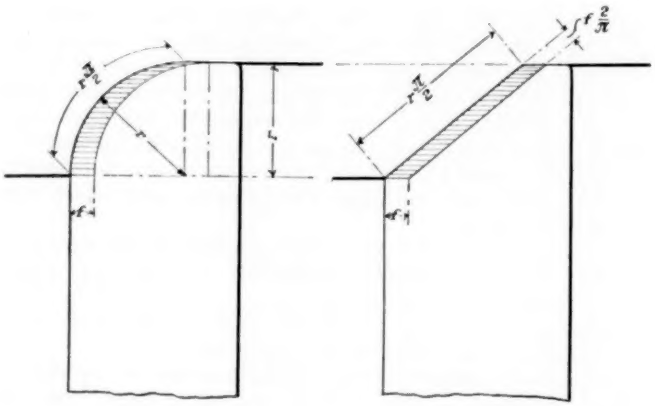

Fig. 11 Fig. 12

value of the ratio d/D in the formula for maximum thickness of chip as given and used by Mr. Parsons. This formula is

$$t_m = 2f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$$

If in this expression we put x for the variable value of d, the corresponding value of D will be $D - 2d + 2x$, whence

$$t_{mx} = 2f\sqrt{\frac{x}{D - 2d + 2x}\left(1 - \frac{x}{D - 2d + 2x}\right)}$$

an expression which for relatively small values of d may, without much error, be simplified to

$$t_{mx} = 2f\sqrt{\frac{x}{D}}$$

This suffices to show that the problem involved in milling is an exceedingly complex. one from a mathematical point of view as soon as we go beyond plain slabbing or rectangular-groove milling. In the writer's complete slide rules for milling he keeps well on

the safe side in estimating the power required for a form cutter by allowing it as much power as a rectangular-groove cutter of a width equal to the maximum width of the form cutter.

Despite the rather adverse criticism of the author's mathematical work, the writer appreciates that he has given us some idea of the cutting resistance of brass and aluminum, to which we have never before seen any reference.

The Author. In reply to the criticism by Messrs. Graves and Hall, of the straight line, Fig. 1, representing hp. output from motor, and the method used to compute hp. to cutting tool, it was recognized that these did not give theoretical accuracy. However, theoretical accuracy was not especially desirable. The methods used in establishing the line and formulas for hp. delivered to cutting tool were convenient approximations and appeared to be more accurate than the actual plotted motor tests, which show some decided humps.

The method proposed in the criticism would change the results by trifling amounts, probably considerably less than unavoidable errors of observation, instruments, etc., and variations in cutter sharpness, and the change would take place at the higher motor powers, where of least effect on the conclusions.

As to machine efficiency, it *is* a difficult matter to establish an efficiency for the combined spindle and table drives, and undoubtedly impossible to establish a figure which does not vary under different cutting conditions. However, the author's experience is that a loss in the efficiency of the one drive is partly, at least, compensated by an increased efficiency of the other, so that the net result over the range of practical conditions is a combined machine efficiency which, for the purpose of the tests shown, may disregard the individual efficiencies. To add the apparently unnecessary complication of extra items involved by considering feed and speed trains separately would defeat the main object of the investigation by making the computations, or the slide rule, too complicated for ordinary use, and remove the tests from the chosen field of actual shop conditions.

As to the objection raised to the assumption that the lines for steel and cast iron meet at a point of A.T.C. = 0.0001 in. and a cubic inch per tool hp-min. of 0.4, undoubtedly this assumption is theoretically incorrect; and if it were possible to increase the sharpness of the cutting edge, etc., in proportion as the chip becomes thinner, the lines would eventually become parallel as the chart Fig. 3 was extended to the left, the lines for the weaker materials being above the stronger materials in all cases. This again, however, becomes merely theoretical, since an A.T.C. of 0.0001 in. is the probable extreme limit of practice, and in fact considerably below the extreme of ordinary practice which is represented by milling cutters with extremely small feeds per minute, fine tooth spacing, and high r.p.m. Such an A.T.C. is not approach-

able by a lathe tool, for instance, and what happens below an A.T.C. of 0.0001 in. is idle speculation.

Regarding the formulas assigned by Taylor as the result of his tests, which are quoted in the criticism, the author is aware that for any given series of tests covering a small range of chip thickness, as did Taylor's, other formulas than those presented in this paper may be made to fit with perhaps even greater apparent accuracy. Taylor, however, does not seem to have recognized the fact which to the author appears from these tests to be fundamental, namely, that it is the *average* thickness of the undistorted chip which should be the basis of all formulas calculated to compute power required when cutting metal, and that on no other basis can the power for cutting metal with different types of tools be compared.

Replying to Mr. Oxford, it was recognized and noted that cutter cooling and lubrication might have an important bearing on the question of power consumed. It was a matter of extreme regret to the author that no data of sufficient worth were available to enable suitable factors to be provided in the formulas. It is to be hoped that this phase of the work can be carried forward by others in the near future.

As regards the number of teeth in spiral milling cutters, although it might appear desirable to use more teeth and an r.p.m. low in relation to feed per minute to obtain power economy comparable with wide-spaced teeth, yet the cutter has higher r.p.m. with slower feeds for a given chip thickness, or in other words, the wide-spaced cutter teeth should give a better finish with equal power economy since finish is determined by revolution marks and not by tooth marks. The wide-spaced cutter would make more revolutions for a given feed distance. In ordinary practice the cutter r.p.m. is established to give a certain standard peripheral speed, without regard for the number or spacing of cutter teeth, and the wider spacing is then more economical of power.

Replying to Mr. Barth, it is to be borne in mind that the method used by the author for eliminating friction and electrical losses was devised primarily to permit tests under the actual conditions of every-day use, the whole purpose of the tests being to get practical results rather than theoretical accuracy. That the lathe and drill tests mentioned in the paper coincide remarkably with the results obtained by the author, seems to show that the methods used by each investigator were reasonably accurate, and within the limits of observation and instrument errors.

The criticism that the author deals only with power consumption needs no answer, since nothing to the contrary was attempted or asserted. There was surely no intention to detract from the remarkable work accomplished by Taylor as regards the life of cutting tools, nor to assert that the laws governing tool life were of less importance than he assigned to them. If Mr. Barth has for eighteen years, and with success, been making slide rules for determining the power required for milling under practical conditions, it is un-

fortunate that they were not more widely known, since a duplication of effort has undoubtedly taken place, and there are so many problems to be solved that such duplication is regrettable.

The author's use of Mr. Taylor's tests does not, as implied, indirectly accept them as being accurate. Having completed the tests relative to milling machines the author was anxious to know how his tests compared with those of others, and those by Taylor seemed best suited for consideration as lathe tests, for the same reasons that those by Messrs. Bird and Fairfield seemed best for comparison with drill tests. It would be more accurate to say that the author unreservedly accepted the tests mentioned as being accurate, and changed the form of Taylor's power equations only because the equations given in Taylor's work offered no possibility of comparison of tests on machines so dissimilar as millers, lathes, and drillers.

The author cannot agree that such acceptance disproves (as asserted) anything advanced in his paper. The comparative range of average chip thickness in the Taylor tests was only 10 to 1, while in the author's miller tests the range covered was 200 to 1 in the case of the cast-iron tests. That a formula which accords well with the tests of both Taylor and the author should be considered disproved because it does not exactly coincide with Taylor's formula which fits the lesser range, is unreasonable.

The author must plead guilty to the charge of using an approximation in the formulas for average chip thickness for milling cutters. However, this was never intended to be misleading, and the statement was plainly made that the formula for A.T.C. was only accurate for practical purposes. The path of the cutting tooth of a slab milling cutter is a complicated curve, only to be computed by the use of the calculus. Such a method of computation would considerably reduce the audience and completely destroy the possibility of ordinary shop use of the formulas. To proceed from this admitted theoretical inaccuracy to the argument that the average undistorted chip thickness is not a true measure of total work done in removing the chip, is another matter. Mr. Barth states that it can be shown that an ideal uniform chip of thickness t_i which requires for its removal the same amount of work as a straight taper chip, may be derived by a formula which he gives. Possibly it can be shown mathematically that this should be true, but in practice the matter is, in addition to other complications, greatly complicated by the fact that the cutter or tool when cutting a wedge-shaped chip does not bite into the metal until the theoretical chip has arrived at a thickness which will vary according to the degree of cutter dulling and the rigidity of the machine. A strictly mathematical solution to determine the thickness of an ideal chip of uniform thickness requiring for its removal the same quantity of power as a wedge-shaped chip, is therefore neither practical nor accurate, and presupposes conditions which cannot exist, especially for the very thin chips.

The author cannot determine where Mr. Barth conceived the idea that the paper presents a general conclusion that the power consumption per cubic inch of metal removed is the same, regardless of the kind of cutting tool used. Indeed, the facts are quite the contrary. The general conclusion borne out by the tests is that the measure of power required is the average thickness of the undistorted chip, irrespective of the tool used, but in obtaining the true average thickness of the chip, allowance must be made for every variation in basic form of tool or method of cutting, as evidenced by the allowance made for the radius or round on lathe tools, and on the corners of face mills, and likewise for formed milling cutters. That power consumption must be greater for a fillet tool or round-nosed lathe tool than for a straight tool, and greater for a formed milling cutter than for a slabbing cutter, is precisely what the author has shown. Due allowance is made for this increased power in the formulas and on the slide rule by reducing the apparent average chip thickness to the true average chip thickness for the respective tools, although as previously stated and for reasons noted, the methods are approximations.

The author's problem was not, as Mr. Barth seems to believe, to obtain mathematical accuracy, which would necessitate adding to the computations all minor factors, but rather, by eliminating those which were of least importance and by approximations of the remaining factors, to obtain expressions which held reasonably true over the practical range of machine use and were capable of being placed in a form conveniently usable. This is what was actually accomplished.

No. 1898

THE EFFICIENCY OF THE SCOTCH MARINE BOILER

By C. J. Jefferson,[1] New York, N. Y.

Member of the Society

The average merchant cargo steamer develops boiler efficiencies of but 55 to 60 per cent with coal-fired boilers, and 60 to 65 per cent with oil-fired boilers. That these efficiencies are much below those possible is shown by the series of tests given in this paper, in which efficiencies as high as 76.7 per cent with coal and 82.8 per cent with oil were developed. Considerable fuel savings are possible in marine practice by means of proper supervision of the operating force.

FROM the analysis of the logs of approximately 250 vessels, it is apparent that the average merchant-marine cargo carrier develops approximately 60 to 65 per cent efficiency in her boiler plant when oil fired, and that these values are from 55 to 60 per cent for coal-fired boilers. This same condition holds true for the average stationary plant of less than 1000 boiler hp., as has been shown by several analyses made at various times of boiler tests conducted on this type of plant.

2 The time has come, however, when better results must be obtained. The Diesel engine is rapidly entering the marine field and the steam-driven vessel must develop its maximum efficiency if it hopes to be able to sail in competition with the Diesel-engined ship, even after making all allowances for the relative difference in the first cost for installation.

3 Moreover, the merchant marine of the United States has established a higher standard of living for its operating personnel than its competitors. This higher standard means increased cost of operation, and this increased cost must be met by increased efficiency of performance.

4 The first essential in obtaining higher efficiencies is to educate the operating personnel to the point where they appreciate what higher efficiencies mean, how to use the instruments necessary for

[1] Head of Fuel Conservation Section, United States Shipping Board.

Presented at a meeting of the Metropolitan Section of The American Society of Mechanical Engineers, New York, November 14, 1922. Slightly abridged.

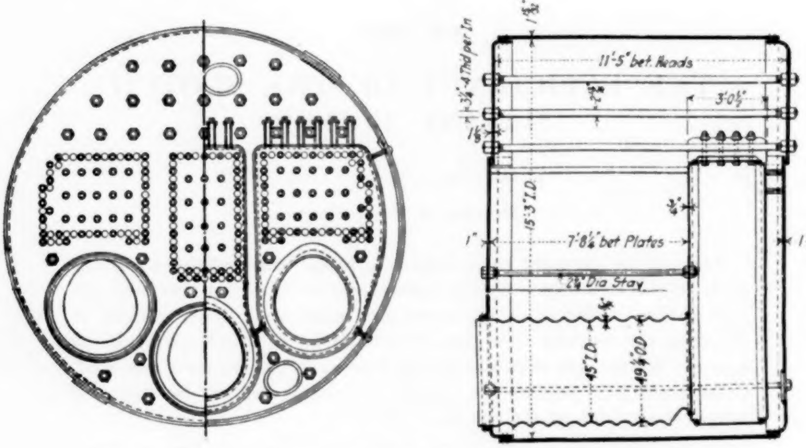

FIG. 1 GENERAL ARRANGEMENT OF SCOTCH MARINE BOILER TESTED

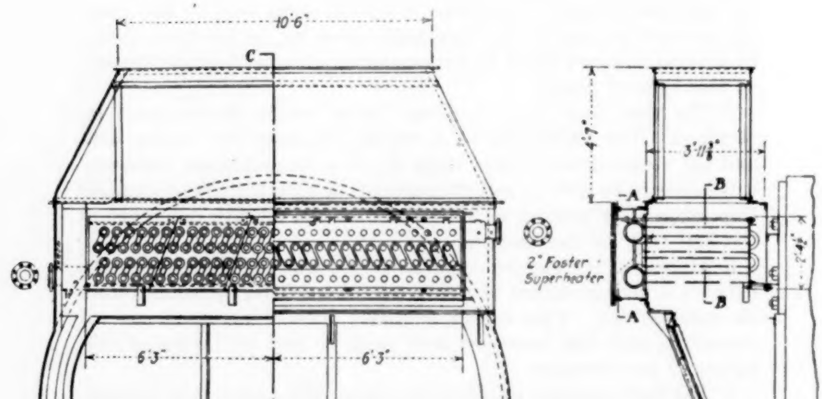

FIG. 2 SUPERHEATER USED WITH SCOTCH MARINE BOILER TESTED

determining the efficiency of combustion, and how to correct faulty combustion and thereby build up boiler efficiency.

5 The tests which are outlined below were conducted by the

TABLE 1 HAND-FIRED COAL TESTS OF SCOTCH MARINE BOILER

Boiler: 3-furnace, with separate combustion chambers; heating surface, 2777 sq. ft. grate surface, 61.8 sq. ft.; area through tubes, 11.94 sq. ft.; combustion chamber volume, 289.4 cu. ft.; heating surface of air heater, 1220 sq. ft.; superheater heating surface, 774 sq. ft.; retarders in tubes; bridge wall, C.I.V.; coal used, Georges Creek run of mine.

Test number	1-1-1-10	1-1-1-9	1-1-1-8	1-1-1-11
Duration, hr.	12.0	10.0	8.08[illegible]	6.089
Total fuel fired, lb.	9,696	12,000	12,[illegible]	11,619
Fuel as fired, lb. per sq. ft. of grate surface per hr.	13.07	19.4	25.1	30.9
Proximate analysis of fuel:				
Moisture per cent	2.15	2.27	1.97	2.17
Fixed carbon, per cent	70.93	69.87	70.00	69.44
Volatile, per cent	18.84	19.07	19.52	19.14
Ash, per cent	8.08	8.79	8.51	9.25
Heating value, B.t.u. per lb.	13,929	13,779	13,858	13,683
Ultimate analysis of fuel:				
Hydrogen, per cent	4.61	4.58	4.58	4.56
Carbon, per cent	80.08	79.34	79.78	79.01
Nitrogen, per cent	1.91	1.89	1.91	1.89
Oxygen, per cent	4.43	4.52	4.27	4.42
Sulphur, per cent	0.89	0.88	0.95	0.87
Ash and refuse:				
Total weight, lb	1200	1340	1309	1125
Per cent referred to dry coal	12.66	11.40	10.60	9.90
Heating value, B.t.u. per lb.	3110	4540	4120	4938
Flue-gas analysis:				
CO_2, per cent	11.2	12.2	12.3	11.8
O_2, per cent	6.8	6.6	6.4	7.3
CO, per cent	0.07	0.2	0.3	0.02
N_2, per cent	81.93	81.0	81.0	80.88
Boiler pressure, gage, lb. per sq. in.	163	157	166	158
Superheat, deg.	1	14	23	38
Moisture in steam leaving boiler, per cent	0.58	0.50	0.55	0.57
Drafts and air pressures (inches of water):				
Below grates	+0.19	+0.54	+0.605	+0.900
Furnace	+0.02	+0.16	+0.283	+0.380
Below air heater	−0.08	−0.058	−0.065	−0.090
Base of stack	−0.12	−0.122	−0.092	−0.130
Gas temperatures (deg. fahr.):				
Leaving boiler	437	503	537	606
Leaving superheater	372	447	472	519
Leaving air heater	277	336	361	396
Total water fed to boiler, lb.	100,560	124,670	122,328	116,988
Feedwater temperature, deg. fahr.	212	204	193	208
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	3.14	4.71	5.78	7.4
Actual evaporation per lb. of coal fired.	10.37	10.40	9.76	10.07
Efficiency of boiler without superheater, per cent	75.2	76.7	72.6	74.6
Heat balance:				
Heat absorbed by boiler, per cent.	75.8	77.9	73.9	76.8
Loss due to moisture in fuel, per cent.	0.2	0.2	0.2	0.2
Loss due to burning hydrogen in fuel, per cent	3.4	3.5	3.5	3.6
Loss due to heat carried away in dry gas, per cent	6.2	7.1	7.4	9.3
Loss due to CO, per cent	0.4	0.9	1.3	0.1
Loss in combustion — ash and refuse, per cent	2.5	3.3	2.8	3.1
Loss unaccounted for, per cent	11.5	7.1	10.9	6.9
Heat in fuel as fired, per cent	100.0	100.0	100.0	100.0

U. S. Shipping Board to obtain reliable and accurate data as to the possible efficiency and capacity of the Scotch marine boiler, which is one of the oldest types of boilers in service and which has

been accepted as a good, reliable servant without due regard being paid to its possible efficiency.

6 The boiler selected for the test purposes was a single-ended, three-furnace, separate-combustion-chamber type, having 2777 sq. ft. of heating surface with coal fire and 3022 sq. ft. with oil

TABLE 2 FUEL-OIL TESTS OF SCOTCH MARINE BOILER — HOWDEN FRONT FORCED DRAFT

Boiler: 3-furnace, with separate combustion chambers; heating surface, 3022.4 sq. ft.; heating surface of air heater, 1220 sq. ft.; superheater heating surface, 774 sq. ft.; retarders in tubes.

Test number	26	36	43
Duration, hr.	8.03	2.888	3.026
Combustion space, cu. ft.	561	545	588
Total fuel fired, lb.	6704	3794	6091
Fuel per burner per hr., lb.	278	438	671
Fuel per hr. per sq. ft. of heating surface, lb.	0.276	0.435	0.666
Analysis of oil as fired:			
Moisture, per cent	0.9	0.3	0.3
Sediment, per cent	trace	trace	trace
Hydrogen, per cent	11.9	11.32	11.6
Carbon, per cent	83.97	82.73	84.5
Sulphur, per cent	3.9	4.0	3.6
Gravity, deg. B	15.4	15.4	15.4
Specific gravity	0.963	0.963	0.963
Flash point, deg. fahr.	166	166	166
Burning point, deg. fahr.	234	234	234
Heating value, B.t.u. per lb.	18,193	18,324	18,234
Flue-gas analysis:			
CO_2, per cent	10.8	12.3	12.6
O_2, per cent	6.3	4.5	3.7
CO, per cent	0.03	0	0
N_2, per cent	82.87	83.2	83.7
Weight of gas per lb. of fuel, lb.	19.37	16.9	16.83
Boiler pressure, gage, lb. per sq. in.	170	172	178
Superheat, deg.	24	32	45
Moisture in steam leaving boiler, per cent	0.65	0.62	0.62
Drafts and air pressures (inches of water):			
Furnace	+ 0.29	+ 0.19	+ 1.28
Below superheater	− 0.08	− 0.09	− 0.05
Below air heater	− 0.11	− 0.11	− 0.09
Base of stack	− 0.14	− 0.14	− 0.11
Gas temperatures (deg. fahr.):			
Leaving boiler	533	597	665
Leaving superheater	470	512	571
Leaving air heater	341	385	428
Total water fed to boiler, lb.	97,653	56,574	88,130
Feedwater temperature, deg. fahr.	188	218	206
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	4.29	6.72	10.10
Actual evaporation, lb. per lb. of oil fired	14.57	14.91	14.47
Efficiency of boiler without superheater, per cent	82.80	81.80	80.73
Heat balance:			
Heat absorbed by boiler and superheater, per cent	84.6	83.8	83.5
Loss due to moisture in fuel, per cent	0	0	0
Loss due to burning hydrogen in fuel, per cent	6.0	5.6	6.0
Loss due to heat carried away in dry gas, per cent	3.8	3.9	4.4
Loss due to CO, per cent	0.2	0	0
Loss in unconsumed oil and unaccounted for, per cent	5.4	6.7	6.1
Heat in fuel as fired, per cent	100.0	100.0	100.0

fire. The boiler was fitted with a Foster waste-heat superheater having 774 sq. ft. of heating surface, this being placed within the gas pass above the smokebox. Above this superheater was an air heater having 1220 sq. ft. of heating surface which heated the air supply to the Howden fronts when running forced draft. Fig. 1

shows the general arrangement of the boiler and Fig. 2 that of the superheater.

7 Tests were divided into four distinct groups, namely, hand-fired coal, pulverized coal, forced-draft oil fire, and induced-draft

TABLE 3 FUEL-OIL TESTS OF SCOTCH MARINE BOILER — NATURAL-DRAFT REGISTERS

(Boiler same as used in tests of Table 2)

Test number	97	94	67	79
Duration, hr.	3.00	2.498	3.203	2.981
Combustion space, cu. ft.	560	560	561	564
Total fuel fired, lb.	2111	2868	4408	4625
Fuel per burner per hr., lb.	235	383	459	517
Fuel per hr. per sq. ft. of heating surface, lb.	0.233	0.380	0.455	0.513
Analysis of oil as fired:				
Moisture, per cent	0	0.1	0.1	0
Sediment, per cent	trace	trace	trace	trace
Hydrogen, per cent	11.37	11.25	11.34	11.78
Carbon, per cent	84.08	83.82	83.60	83.98
Sulphur, per cent	4.0	3.9	3.5	3.8
Gravity, deg. B.	15.4	15.4	15.4	15.4
Specific gravity	0.963	0.963	0.963	0.963
Flash point, deg. fahr.	166	166	166	166
Burning point, deg. fahr.	254	254	254	254
Heating value, B.t.u. per lb.	18,382	18,418	18,248	18,459
Flue-gas analysis:				
CO_2, per cent	12.8	13.06	12.2	11.98
O_2, per cent	3.3	3.0	4.4	4.4
CO, per cent	0.03	0.17	0.01	0.06
N_2, per cent	84.87	83.77	83.39	83.56
Weight of gas per lb. of fuel, lb.	16.48	15.95	17.19	17.49
Boiler pressure, gage, lb. per sq. in.	172	174	172	168
Superheat, deg.	0	14	28	33
Moisture in steam leaving boiler, per cent	0.62	0.64	0.62	0.47
Drafts and air pressures (inches of water):				
Furnace	−0.14	−0.22	−0.31	−0.64
Below superheater	−0.15	−0.25	−0.48	−0.78
Base of stack	−0.15	−0.31	−0.57	−0.84
Gas temperatures (deg. fahr.):				
Leaving boiler	448	525	589	548
Leaving superheater	420	473	503	490
Total water fed to boiler, lb.	32,276	41,282	61,813	66,348
Feedwater temperature, deg. fahr.	232	212	191	212
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	3.63	5.69	6.79	7.68
Actual evaporation, lb. per lb. of oil fired	15.29	14.39	14.02	14.35
Efficiency of boiler without superheater, per cent	82.35	78.98	79.29	78.70
Smoke, Ringlemann scale	1.5	1.75	1.5	2.5
Heat balance:				
Heat absorbed by boiler and superheater, per cent	82.9	80.2	81.1	80.6
Loss due to moisture in fuel, per cent	0	0	0	0
Loss due to burning hydrogen in fuel, per cent	5.7	5.7	6.0	6.1
Loss due to heat carried away in dry gas, per cent	7.1	8.4	9.6	9.4
Loss due to CO, per cent	0.1	0.6	0	0.2
Loss in unconsumed oil and unaccounted for, per cent	4.2	5.1	3.3	3.7
Heat in fuel as fired, per cent	100.0	100.0	100.0	100.0

oil fire. The pulverized-coal experiments were conducted to the point where it was demonstrated that this type of firing is not feasible for marine service with the Scotch boiler, as the limited size and water-cooled feature of the furnaces and combustion

chambers prevented operating at ratings which were sufficiently high to meet the demand made by the average marine boiler plant.

8 Four of the coal-burning tests which show typical results are given in Table 1. These tests are selected as representing results covering a combustion range of from 13.07 lb. to 30.9 lb.

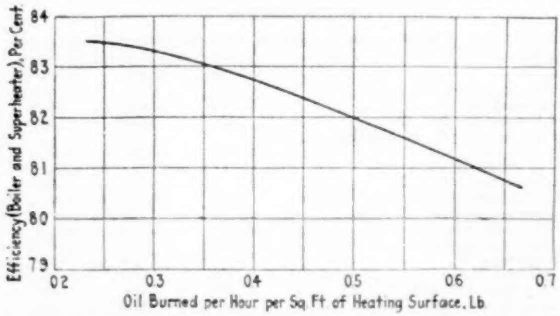

Fig. 3 Average Combined Efficiency Values Obtained in the Forced-Draft Oil-Burning Tests

(1002.46 × lb. oil per sq. ft. of heating surface = oil per burner.)

of coal per hour per square foot of grate surface. It will be noted that the plant efficiency including boiler, superheater, and air heater ranges from 73.9 to 77.9 per cent. All of these tests were run under forced-draft operating conditions.

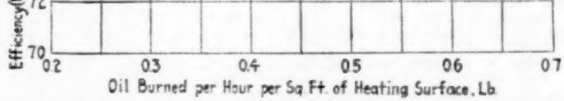

Fig. 4 Average Combined Efficiency Values Obtained in the Induced-Draft Tests

(1002.46 × lb. oil per sq. ft. of heating surface = oil per burner.)

9 The forced-draft oil-burning tests were conducted using Dahl, Schutte & Koerting, Coen, Todd, and Bethlehem Shipbuilding Corporation burners, all of these except the last named being used in conjunction with the Howden combined oil- and coal-burning front.

10 Table 2 gives the results for three representative tests of the forced-draft oil-fired series covering a range of from 0.276 to 0.666 lb. of oil per hour per square foot of heating surface. It will be noted that the combined boiler, superheater and air heater efficiency ranges from 83.5 to 84.6 per cent. Fig. 3 gives the average efficiency values covering the entire series of forced-draft oil-burning tests.

11 The induced-draft tests were conducted with the Bethlehem Shipbuilding Corporation burner as well as with the Schutte & Koerting types "L" (modified) and "N," the Coen, the Todd, and the Enco burners. This latter burner was originated at the Philadelphia Navy Yard Fuel Oil Test Plant, and its adaptation for use in Scotch boiler service was developed during these tests.

12 The ratings obtained in these induced-draft tests were limited by the amount of available draft as the height of stack above the center of furnace was only 40 ft., and this draft was augmented by a ½-in. steam nozzle. The combined effect of the stack and nozzle produced a 0.9-in. draft at the base of the stack. Table 3 gives the data of four representative tests covering a range of from 0.233 to 0.513 lb. of oil per hour per square foot of heating surface and showing efficiencies ranging from 80.2 to 82.9 per cent. Fig. 4 gives the average efficiency values obtained from all of the tests of this class.

13 In comparing Figs. 3 and 4, the fact must be kept in mind that the induced-draft tests did not have the help of the air heater in building up furnace efficiency.

14 From the foregoing data it will be seen that the Scotch marine boiler is capable of efficiencies considerably in excess of the average operating values obtained, and it is, therefore, a matter of simple calculation to show what considerable fuel savings are possible when reasonable supervision is shown on the part of the operating force.

DISCUSSION

Charles D. Shepard.[1] While the tests cited in Mr. Jefferson's paper show very good results, there are several points wherein they differ from marine practice.

One of the most noticeable things in the tests is the variation in the average feedwater temperatures, which ranged from 188 to 212 deg. fahr., with one temperature of 232 deg. It is interesting to note that the highest boiler efficiency was obtained with the lowest average feedwater temperature used in any of the runs. Had a higher feedwater temperature been used in this run the boiler efficiency would undoubtedly have been higher, as by a rule of thumb a saving of one per cent in fuel is obtained for each increase of 10 deg. fahr. in the temperature of the feedwater. This is

[1] Washington, D. C.

one of the things about which marine engineers are particularly careful, and temperatures of from 220 to 230 deg. fahr. are the usual practice, and are obtained by having a back pressure of between 2 to 6 lb. gage on the auxiliary exhaust which is used to heat up the feedwater. Higher temperatures up to 240 deg. are sometimes used, but any temperature requiring an auxiliary exhaust pressure of more than 10 lb. gage is not good practice. By carrying a high feedwater temperature there is a saving in fuel, and this temperature is easily read with a thermometer. The adjustment of the valves to carry the proper auxiliary feed pressure for the desired feedwater temperature is readily made and need not be changed ordinarily under constant working conditions.

The introduction of flue-gas analysis has been slow in marine work, probably due to the idea that it is a complicated process. It also requires chemicals of known strength, which are not readily obtainable everywhere, and some of them are liable to deteriorate if not kept properly. It is true that flue-gas analysis requires a certain knack to obtain proper results, but given a few simple instructions, it would seem that marine engineers would quickly learn to make accurate readings. After using this method to learn what is going on in the furnace, and making the required changes in the operation of the boilers to suit such an analysis, a very real saving in fuel would result in a large number of cases. The use of automatic CO_2 and CO recording instruments has not been possible, due probably to the delicacy of the instruments, which are so devised that they can be used only for stationary work.

Pyrometers are used in navy work to indicate the temperature of the gases as they leave the boiler. It seems that they would be equally valuable in merchant-marine work, and would afford the engineer some indication of how much heat is going up the stack and being wasted.

The marine engineer is also handicapped by not having a definite means provided for measuring the amount of water fed to the boiler, and when using coal, for measuring the exact amount fired. Approximations may be used in both cases — for example, so many pounds of coal in a wheelbarrow — and with a well-made-up system of piping and no free escapes of steam the amount of make-up feed used is a fairly definite proportion of the water fed to the boilers. Also for a complete analysis of boiler conditions an instrument for determining the quality of the steam, if not superheated, would be required.

A further important thing in marine work is the use of a calorimeter for determining the fuel value of the coal or oil used. This would give the engineer an idea of the quality of the fuel he was using, and would also be of great value in its purchase. The recent high price of coal and its poor quality will undoubtedly hasten the use of a calorimeter for this work.

Such items as keeping the boiler clean by opening at the proper intervals, blowing the tubes regularly, keeping the boiler water

slightly alkaline and not too salty, and all apparatus in good working condition, are too well known to need any discussion.

These, then, are the principal items in which it seems to the writer that this test differs from ordinary present merchant-marine practice. In addition he does not himself know of any case in marine work where the gases leaving the furnace are used to heat up the air fed to the boiler by means of an air heater separate from the boiler, as he understands was the case in this test installation; but of course this system is very valuable as far as the boiler efficiency is concerned.

The Author. In regard to Mr. Shepard's comments the author submits the following:

Feedwater Temperature. All the efficiencies as listed in the heat balance of this test were determined from the equivalent evaporation figures which automatically take care of the feedwater temperature.

The writer has found it common practice in marine service to operate with 15 lb. back pressure, and on several of our best performers we have gone still further and installed a second feed heater which utilizes the auxiliary exhaust from such auxiliaries as are run on full boiler pressure, these auxiliaries being run at 35 lb. back pressure. The purpose of the second feed heater using the higher back pressure is to boost up the feed temperature to about 270 deg. This has resulted in increased economies as high as 7 per cent of steam for our turbine-driven ships where the auxiliary steam being taken away from the main condenser permitted higher vacuums to be obtained on the main unit and the increased boiler-feed temperature resulted in reduced maintenance charges.

Flue-Gas Analysis. Mr. Shepard comments regarding the fact that the adoption of flue-gas analysis for the use of determining boiler efficiencies in marine practice has been somewhat slow. You are advised that we have found this only too true, and it is to overcome such conditions as this that the Fuel Oil School has been started at the Philadelphia Navy Yard. It is proper to say, however, that while there were practically no Orsats in use in our fleet a year ago, now practically 50 per cent of the vessels are equipped with this instrument and the *engineers are using them.*

Use of Pyrometers. The pyrometer has been installed for some time in merchant vessels, but in common with a great many other similar power-plant installations, proper regard has not been given to its location. In order to get really worth-while data for operating purposes, it is necessary that the pyrometer be installed as close as possible to the point where the gases leave the boiler. This in the Scotch boiler really means, on a three-furnace job, three pyrometers. When the pyrometer is installed in the stack halfway up the fiddley, the temperatures observed are not trustworthy for determining efficiency of operation as this temperature

is generally affected considerably by minor leakage through breachings and uptakes.

Use of the Fuel Calorimeter. The author's personal experience with the fuel calorimeter forces him to the conclusion that this is a laboratory instrument, and if used by inexperienced persons would lead to very erroneous conclusions.

Use of Air Heaters. Mr. Shepard states that he knows of no case where the gases leaving the furnaces are used to heat up the air fed to the boiler. This system of air heater was developed by James Howden a number of years ago and is in use on practically all Scotch boiler installations, as well as the majority of the large passenger vessels such as the *Leviathan*, etc.

No. 1899

THE INTEGRATING GATE: A DEVICE FOR GAGING IN OPEN CHANNELS

By H. E. Doolittle,[1] San Diego, Cal.

Non-Member

The integrating gate is a device for measuring the flow of liquids in open channels which, it is claimed, avoids some of the disadvantages of present methods, such as the weir, the rating flume, the venturi flume, etc. It consists of a freely swinging gate suspended in the channel and practically closing it. The flow of liquid causes this gate to assume an angle from the vertical, which angle is an index of the flow. Sufficient experimentation has been done to show that the error and individuality of different gates is small enough for practical purposes. The rating curve can be predetermined, and the loss of head through the gate is considerably less than in the weir or venturi flume.

THE common methods of gaging flow in open channels, including the weir, the rating flume, and the venturi flume, have certain disadvantages, among which may be mentioned accumulations of rubbish and silt which affect the relations of quantity and head, restriction of the flow channel, and lack of direct-reading qualities.

2 The integrating gate, which is largely free from these objections, consists of a rectangular plate hung from a horizontal axis at right angles to the direction of flow in such a manner that when no water is flowing the gate is vertical, practically filling the channel section. The water pushes the gate open slightly and flows under it (see Fig. 1). The angle between the plane of the gate and a vertical plane through its axis is a function of the quantity of water flowing.

3 The advantages of such a device may be enumerated as follows: It is simple and inexpensive to construct, and simple in operation. It has but one working part, and should require little maintenance. It is sufficiently accurate to meet practical needs, and should be unaffected by silt as the velocity on the bottom of

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Abstract of paper awarded A.S.M.E. Second Student Prize at the Annual Meeting, New York, N. Y., December 3 to 6, 1923, of The American Society of Mechanical Engineers.

the channel is greatly increased in the vicinity of the gate. It has less than half the loss of head of the corresponding weir or venturi flume. It has a wide range in discharge capacity. It is direct reading and can be equipped with a suitably calibrated scale to read in any units desired.

Laboratory Equipment and Procedure

4 The arrangement of the laboratory in which the integrating gate was developed is shown in Fig. 2, and a drawing of the flume, gate, and tanks is given in Fig. 3. The apparatus used consisted of:

a A deep-well pump and a 4-in. centrifugal pump, both pumping from a supply tank and giving a combined flow of about 2.9 sec-ft.

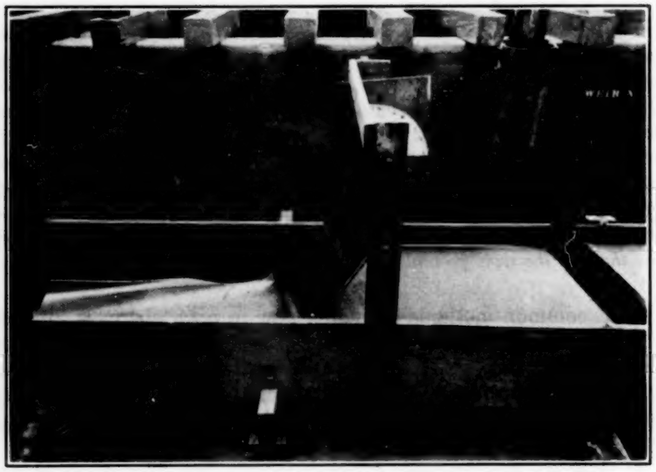


Fig. 1 Gate No. 4, Position No. 2. Trap Open and No Back Wave Forms. Clearance Leakage at *A* Negligible

b A smooth redwood channel 1.766 ft. by 0.91 ft. in cross-section and 23.4 ft. in overall length. Three vertical lath baffles at the intake end smoothed out waves from the pipes discharging into the channel, and a trap at the downstream end allowed the water to discharge either to the supply tank or to a measuring tank. A baffle board was placed across the channel about 1.5 ft. above the gate axis to smooth out surface ripples, and extended about ½ in. below the surface of the water.

c A volumetric means of measuring flow, comprising a measuring tank calibrated to contain 1.524 cu. ft. per cm. rise as indicated by a hook gage graduated in centimeters, and an Equity watch for indicating the length of time of rise.

d An integrating gate, consisting of a metal plate hung at right angles to the channel on two tool-steel pins ¾ by $\frac{5}{64}$ in. in

diameter, fitted into holes drilled in the edges of the plates and into the ends of supporting stud bolts carried on wooden supporting members. A clearance of about $\frac{1}{8}$ in. was allowed between the gate and the perimeter of the channel. A pointer attached to the gate indicated on a quadrant reading to 15 minutes of arc the departure of the gate from the vertical.

5 The following quantities were measured:

a Hook-gage readings, R_1 and R_2, in the measuring tank at the

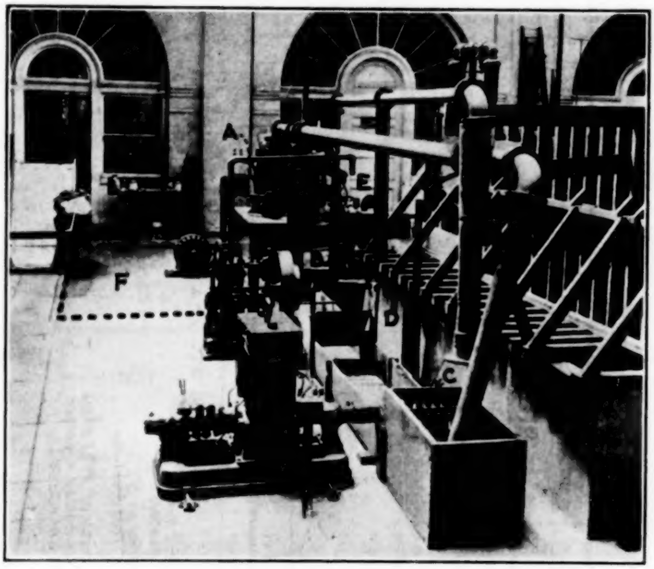


Fig. 2 Showing General Arrangement of Apparatus

A, *B*, and *J*, deep-well pump
B, 4-in. centrifugal pump
C, Intake of channel showing 3 vertical lath baffle plates
D, Integrating gate propped open. Position No. 2
E, Hook gage of measuring tank No. 3
F, The dotted line indicates the position of supply tank No. 4 beneath the wooden flooring.

beginning and end of each run. These were read to the nearest 0.05 cm.

b Time, t_1 and t_2, in minutes and seconds at the beginning and end of each run. The trap acted rapidly enough to permit time to be read to the nearest half-second.

c The angle A of the gate as indicated by the pointer on the quadrant as follows: A_1 at the beginning of the run; A_2 after the water had been pumped back into the supply tank; A_3 at the end of the run with the measuring tank filled and the supply tank lower than at the beginning of the run.

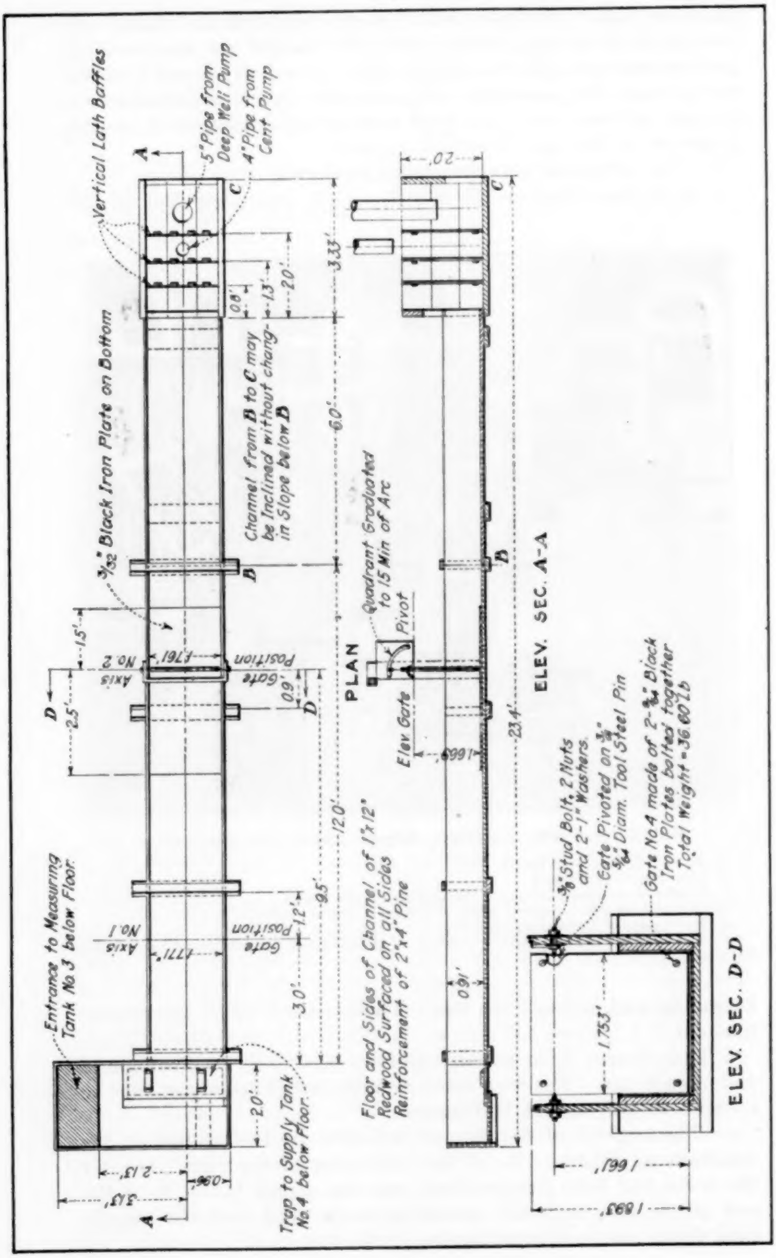


FIG. 3 DIAGRAM OF LABORATORY INSTALLATION OF INTEGRATING GATE

d Amplitude of oscillations, dA, in minutes of arc, due to waves and irregularities of flow.

e Depth of water in the channel, H_A, about 1 ft. upstream from the gate axis, as determined by a rule graduated to 0.01 ft., and read to the nearest 0.005 ft. H_A was measured at the beginning of a run.

f Depth of water in the channel, C, about 7 ft. below the second position of the gate, measured when the trap was down and the water discharging into the measuring tank. This was found to be practically the same with the gate in position and

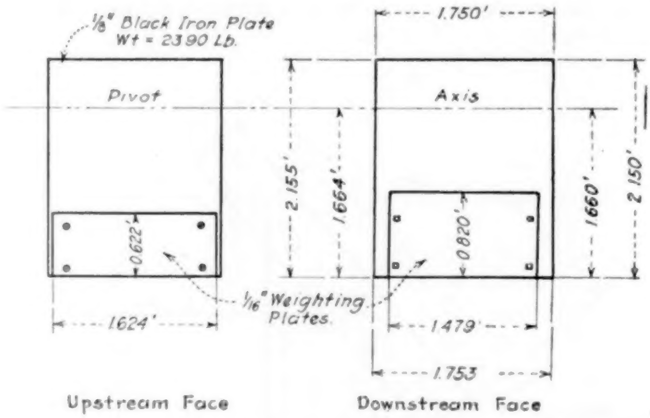


FIG. 4 GATE No. 2

with the gate removed. C was measured by the same means as H_A.

g Distance, E, of back wave below gate from bottom edge of gate. Measured to the nearest 0.1 ft.

h Angle of gate, A_G when $Q = 0$ when hanging by gravity, and A_v with upstream face vertical, as determined by square and level.

6 The gate was set in two different positions. The first was at a point 3.0 ft. and the second at a point 9.5 ft. upstream from the trap. The second position differed from the first in that a smooth, level iron plate on the bottom of the channel eliminated the effect of the warping of the wooden bottom in position No. 1, and the width of the channel was about 0.01 ft. less. The downstream velocities were less due to the increased length of channel below the gate, but the channel intake was considerably nearer, increasing the effect of eddy currents.

7 Different weights of and slightly different sizes of gates were used as follows:

Gate No. 1. Same as gate No. 2 (see Fig. 4) with the weighting

plates removed. The gate was a somewhat rusty iron plate, $\frac{1}{8}$ in. thick with square milled edges. It weighed 23.9 lb., and had a computed static moment, M, about the axis of 14.05 lb-ft. The static moment as determined by supporting the edge on scales was 13.45 lb-ft.

Gate No. 2. Same as gate No. 1, with weighting plates of $\frac{1}{16}$-in. galvanized iron attached. Combined thickness at bottom edge, $\frac{1}{4}$ in. The weight of the downstream plate was 3.20 lb., and that of the upstream plate was 2.80 lb. The static moment (computed) was 21.80 lb-ft., and by experimental determination was 21.34 lb-ft.

Gate No. 3. Same as gate No. 4, with the weighting plates of

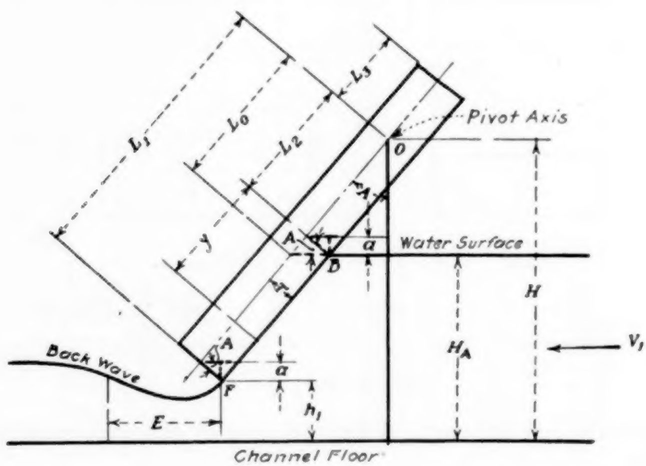


FIG. 5 NOTATION DIAGRAM

gate No. 2 attached above axis to give the same static moment as gate No. 2. Counterbalances were as follows:

Distance, axis to center of gravity, ft.	0.475	0.572	0.31	0.40
Weight, lb.	2.80	3.20	1.55	0.60
Moment, lb-ft.	1.33	1.83	0.48	0.24

The total combined static moment of gate No. 3 was 26.15 − 3.88 = 22.27 lb-ft. Due to errors in computation, this value does not check as closely with the static moment of gate No. 2 as had been planned.

Gate No. 4. Consisted of two pieces of $\frac{9}{64}$-in. black plate iron, bolted together so shown in the sectional elevation *D–D*, Fig. 3. The edges were left as they came from the power shear and were comparatively square. The surface was oiled to prevent

rusting and was very smooth. The gate weighed 36.60 lb., and the computed static moment was 26.15 lb-ft.

8 A particular gate and a particular set of channel conditions constituted a setting. The settings used were as follows:

Setting No. 1. Gate No. 1 in position No. 1. Channel level. One set of runs was made, the gate readjusted for clearance and alignment, the quadrant readjusted, and a second set of runs made to check the accuracy of the readjustments.

Setting No. 2. Gate No. 1, position No. 1. Various slopes were obtained by tilting the entire channel. As it later became apparent that the best results were obtained with a zero slope under the gate, the data from this setting were not used.

Setting No. 3. Gate No. 2, position No. 1. Although the average slope under the gate was zero, actual measurement by a

TABLE 1 TABULATION OF DATA AND COMPUTATIONS

Run No.	A_1 A_2 deg.	A_2 deg.	dA deg.	R_1 R_2 cm.	t_1 t_2 min.	H_A ft.	A deg.	Q sec-ft.
SETTING No. 1: Gate No. 1, Position No. 1, Channel Level.								
FIRST ADJUSTMENT								
1	10 45	Grav.		0	0	0	00 00	0
	10 55	Vert.		0	0			
2	51 22	50 45	45	60.00	36:30	0.730	40 06	2.845
	51 15			228.00	38:00			
3	49 30	48 07	30	44.00	11:00	0.695	38 53	2.455
	49 30			285.60	13:30			
4	47 00	46 45	45	11.45	6:45	0.650	36 06	2.110
	47 10			282.00	10:00			
5	45 07	43 52		21.20	35:15	0.610	33 34	1.814
	45 07			275.00	38:45			
6	41 50	41 22		38.00	42:30	0.550	30 40	1.448
	41 45			265.70	46:30			
7	40 30	39 30		40.00	55:30	0.525	29 03	1.286
	40 22			267.80	60:00			
	35 15	35 00	15	126.30	44:15	0.435	24 12	0.857
	35 15			261.10	48:15			
9	28 22	28 15		24.90	03:00	0.320	17 23	0.423
				124.70	09:00			
10	20 20	20 30		124.80	23:30	0.205	9 30	0.139
				185.00	34:30			

2.5-ft. straight edge and carpenter's level gave values of channel slope at various points which ranged from 0.005 ft. downstream to 0.003 ft. upstream in 2.5 ft.

Setting No. 4. Gate No. 2, position No. 2. Channel slope zero under gate due to iron plate. Channel practically level above gate.

Setting No. 5. Gate No. 2, position No. 2. Channel slope zero under gate. Sine of angle of channel slope from point 2.5 ft. to 8.5 ft. upstream from gate was 0.012.

Setting No. 6. Gate No. 3, position No. 2. Channel slope zero both under and above gate.

Setting No. 7. Gate No. 4, position No. 2. Channel slope zero both under and above gate.

9 Table 1 is typical of the manner in which the data of the various runs were tabulated.

Sample Computations and Theoretical Formulas

10 *Notation* (see also Fig. 5):

A = angle between gate and its initial vertical position, deg. and min.

A_G = angle of the gate when $Q = 0$, gate hanging by gravity

A_v = angle of the gate with its upstream face vertical

A_1 = angle of gate as indicated by the pointer at the beginning of the run

A_2 = angle of the gate after the water had been pumped back into the supply tank

A_3 = angle of the gate at the end of the run, when the measuring tank was filled and the supply tank was lower than at the beginning of the run

a = vertical projection of $(L_0 - L_2)$, ft.

C = depth of water in channel 7 ft. below gate in position 2, ft., measured with trap down and water discharging to measuring tank

$C_K = M$ (sin $A)/M_w$ where M_w was determined from computed values of H_A

$C_M = M$ (sin $A)/M_w$ where M_w was determined from experimental values of H_A

$C_Q = Q/Q$

D = width of channel, ft.

d = width of gate, ft.

dA = maximum oscillation of pointer, min. of arc

E = distance of back wave from bottom edge of gate, ft.

F = loss of head in a venturi flume, ft.

G = total normal force of static head acting on upstream face of gate, lb.

g = acceleration due to gravity

H = vertical distance of gate axis above channel floor, ft.

h = head on center of orifice, ft.

h_1 = height of opening under gate, ft.

H_A = depth of water in channel about 1 ft. upstream from gate axis, ft.

H_w = loss of head in a suppressed weir with tailwater level with crest

L_0 = distance from gate axis to intersection of water surface with center line of gate, ft.

L_1 = distance from gate axis to lower edge, ft.

L_2 = distance from gate axis to intersection of water surface with upstream surface of gate, ft.

L_3 = distance from gate axis to upper edge of gate, ft.

M = static moment of gate about its axis when $A = 90$ deg., lb-ft.

M_w = moment about gate axis of static head on upstream face of gate, integrated over gate surface, ft-lb.

Q = flow through channel, sec-ft.
Q' = flow under gate, computed by orifice formula, sec-ft.
R_1, R_2 = hook-gage readings at beginning and end of run, cm.
t_1, t_2 = time at beginning and end of run, sec.
T = one-half thickness of bottom edge of gate, ft.
V_1 = velocity of approach, ft. per sec.
y = distance from surface of water, along upstream face of gate, to center of pressure of static head, ft.

11 *Sample Computation of Angle "A":*

$$A = \{(A_1 + A_3)/2 + A_2\} \tfrac{1}{2} - A_v \quad . \; . \; . \; . \; . \; . \quad [1]$$

A_1 and A_3 are averaged together first, as they refer to the same conditions of flow. A_2 refers to minimum flow for the run and is averaged with the averages of A_1 and A_2, since a straight-line variation of flow is assumed to occur from the beginning to the end of

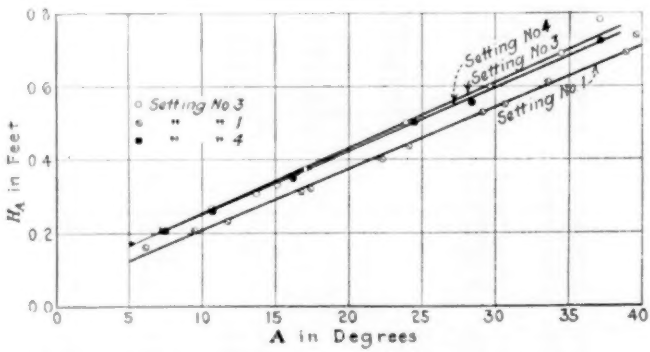


FIG. 6 VARIATION OF DEPTH OF WATER H_A WITH ANGLE A

the run. A_v differs from A_G by not over 10 min. of arc, due to pivoting holes not being exactly in the center of the edges of the gate. From Table 1, runs 1 and 2, A_1 = 51 deg. 22 min., A_2 = 50 deg. 45 min., A_3 = 51 deg. 15 min., A_v = 10 deg. 55 min.

$$A = \tfrac{1}{2}\{(51° \, 22' + 51° \, 15')/2 + 51° \, 45'\} - 10° \, 55' = 40° \, 06'$$

12 *Sample Computation of Quantity "Q."* Since 1.524 cm. rise in measuring tank was equivalent to 1 cu. ft.,

$$Q = 1.524(R_2 - R_1)/(t_2 - t_1) \quad . \; . \; . \; . \; . \; . \quad [2]$$

From Table 1, run 2, R_1 = 60.00 cm., R_2 = 228.00 cm., t_1 = 36:30, t_2 = 38:00, and $(t_2 - t_1)$ = 90 sec., whence

$$Q = 1.524(228.0 - 60.0)/90 = 2.845 \text{ cu. ft. per sec.}$$

13 *Sample Computation of Static Moment "M":*

$$M = (L_2^1 - L_2^3)W/2(L_1 + L_3) \quad . \; . \; . \; . \; . \; . \quad [3]$$

For gate No. 4, Fig. 3 gives $L_1 = 1.661$ ft., $(L_1 + L_3) = 1.893$ ft., $L_3 = 0.232$ ft. W = weight of gate = 36.60 lb.

$$M = \{(1.661^2 - 0.232^2)36.60\}/(2 \times 1.8930) = 26.15 \text{ lb-ft.}$$

14 *Relation of "H_A" and Angle "A."* Given H_A, M_w may be computed by integrating over the submerged area of the gate the moment about the gate axis of the static head of the water acting on the upstream surface of the gate. M_w may then be equated to M (sin A) by introducing a constant, and M (sin A) $= C_M M_w$.

TABLE 2 VALUES OF C_M, C_Q AND C FOR VARIOUS SETTINGS

A deg.	Q sec-ft. (From Fig. 8)	H_A ft. (From Fig. 6)	C_M	C_Q	H_A (Computed)	C_K
			SETTING NO. 4			
10 00	0.150	0.246	0.893	0.695	0.288	0.634
12 00	0.205	0.281	0.889	0.670	0.305	0.736
15 00	0.313	0.330	0.885	0.646	0.337	0.840
20 00	0.583	0.420	0.855	0.641	0.420	0.855
25 00	0.953	0.505	0.838	0.640	0.504	0.841
30 00	1.448	0.591	0.845	0.638	0.585	0.869
35 00	2.070	0.679	0.855	0.636	0.670	0.910
38 00	2.515	0.730	0.894	0.644	0.735	0.871
			SETTING NO. 7			
10 00	0.143				0.266	0.936
12 00	0.201				0.296	0.969
15 00	0.319				0.350	0.919
20 00	0.590				0.431	0.955
25 00	0.977				0.519	0.957
30 00	1.483				0.619	0.968
35 00	2.125				0.708	0.914
38 00	2.580				0.770	0.904
			SETTING NO. 1			
10 00	0.146	0.210		0.719	0.273	0.454
12 00	0.202	0.245		0.692	0.280	0.563
15 00	0.311	0.295		0.670	0.316	0.596
20 00	0.568	0.379		0.650	0.387	0.641
24 30	0.885	0.455		0.642	0.442	0.719
28 00	1.190	0.512		0.637	0.501	0.726
35 00	2.020	0.630		0.641	0.624	0.750
38 00	2.460	0.681		0.648	0.687	0.722

15 In Fig. 5, taking B as the origin, the distance to the center of pressure of the static head acting on the gate is

$$y = \tfrac{2}{3}(L_1 - L_2)$$

The total force of the static head acting on the gate and normal to it is

$$G = d(L_1 - L_2)(L_1 \cos A + a - L_0 \cos A)62.4/2$$

The moment about the gate axis at O is

$$\begin{aligned} M_w &= F(y + L_2) \\ &= d(L_1 - L_2)(L_1 \cos A + a - L_0 \cos A)(\tfrac{2}{3}L_1 - \tfrac{2}{3}L_2 + L_2)62.4/2 \end{aligned}$$

$$L_2 = L_0 - \frac{a}{\cos A}$$

$$M_w = d\left(L_1 - L_0 + \frac{a}{\cos A}\right)\left(L_1 + \frac{a}{\cos A} - L_0\right)\cos A$$
$$\times \left(2L_1 + L_0 - \frac{a}{\cos A}\right)\frac{62.4}{6}$$

and since $L_0 = (H - H_A)/\cos A$ and $a = T \sin A$,

$$M_w = \frac{62.4d}{6 \cos^2 A}(L_1 \cos A - H + H_A + T \sin A)^2$$
$$\times (2L_1 \cos A + H - H_A - T \sin A) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

$$M \sin A = C_M M_w = \frac{10.4dC_M}{\cos^2 A}(H_A - H + L_1 \cos A + T \sin A)^2$$
$$\times (H - H_A + 2L_1 \cos A - T \sin A) \quad . \quad . \quad . \quad . \quad [5]$$

16 It would seem that a correction for velocity of approach

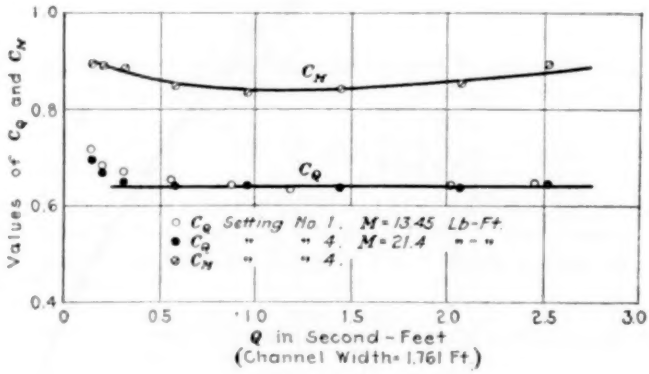


FIG. 7 VARIATION OF C_Q AND C_M WITH Q

should be applied to H_A. The velocity directly back of the gate and parallel to it is evidently very small, since adding a correction of $V_1^2/2g$ to H_A increased M_w so that C_M departed much further from a constant than when this correction was omitted.

17 *Sample Computation of "C_M."* From an assumed angle $A = 30$ deg., and the data of setting No. 4, we have: $\sin A = 0.500$; $\cos A = 0.866$; $\cos^2 A = 0.750$; $M = 21.4$ lb-ft.; $L_1 = 1.662$ ft.; $T = 0.01$ ft.; $d = 1.752$ ft.; $H = 1.670$ ft.; and from Fig. 6, $H_A = 0.591$. Substituting these values in Equation [5] we find a value of

$$C_M = 21.4 \times 0.500/12.78 = 0.845$$

Values of C_M computed in the above manner are given in Table 2, while Fig. 7 gives values of C_M plotted against Q.

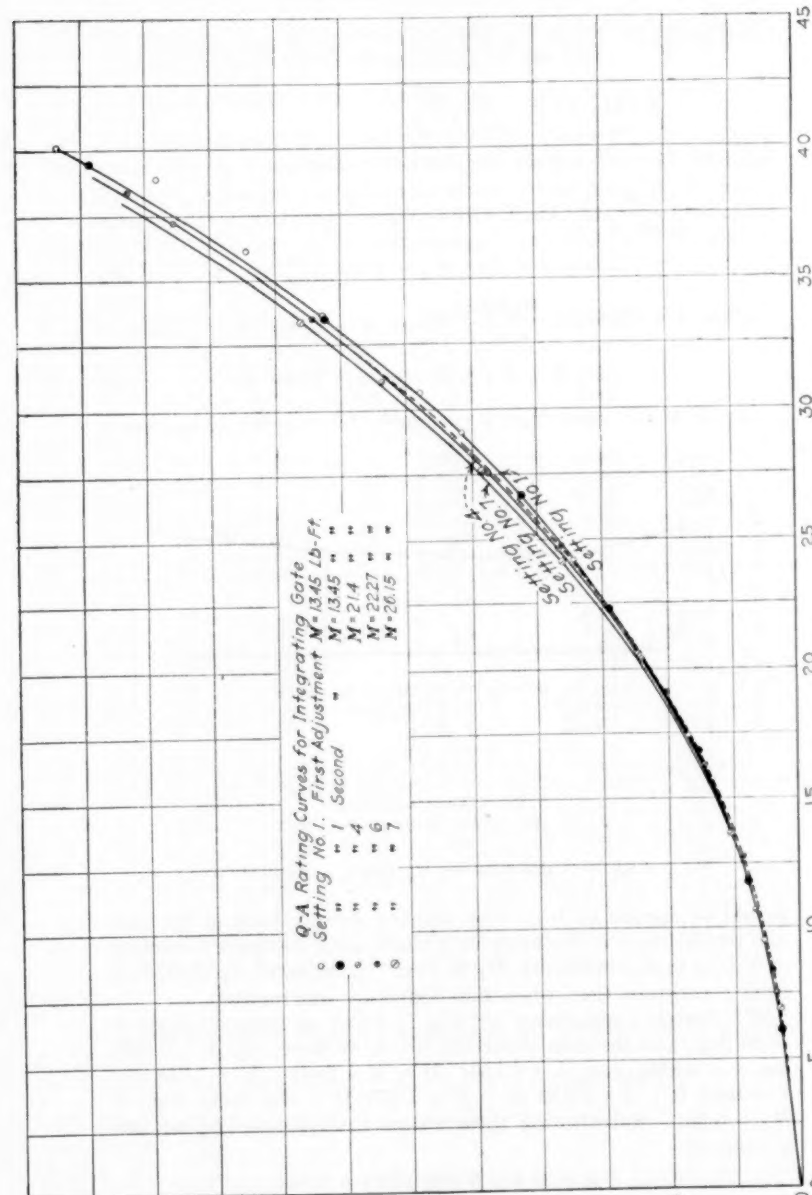


FIG. 8 VARIATION OF QUANTITY Q WITH GATE ANGLE A

18 *Derivation of the Relation between "Q" and "H_A."* Consider the flow under the gate as being produced by the square root of the head on the center of the opening under the gate. From the ordinary orifice formula, $Q = 8.02\ A\sqrt{h}$. This formula is used as it gives more constant values for C_Q than does the precise formula for rectangular orifices. Using this formula, and referring to Fig. 5, we find the head on the center of the orifice to be

$$h = \tfrac{1}{2}\{H_A + L_1 \cos A + a - (H - H_A)\} + \frac{V_1^2}{2g}$$

The area of the opening under the gate is

$$A_A = \{H - (L_1 \cos A + a)\}D, \text{ and } a = T \sin A$$

hence the discharge is

$$Q' = 8.02D(H - L_1 \cos A - T \sin A) \left\{ H_A - \tfrac{1}{2}(H - L_1 \cos A - T \sin A) + \frac{V_2^1}{2g} \right\}^{\frac{1}{2}} \quad \text{[6]}$$

The application of this formula may be seen by using the data in Par. 17, and taking $D = 1.761$, with $Q = 1.448$ sec-ft. (from Fig. 8).

$$\frac{V_1^2}{2g} = \frac{Q^2}{H_A^2 D^2 2g} = 1.148^2/\{(0.591)^2\ (1.761)^2\ 2g\} = 0.0303 \text{ ft.}$$

Substituting these values in Equation [**6**] we find $Q' = 2.27$ sec-ft., and $C_Q = Q/Q' = 1.448/2.27 = 0.638$. Table 2 gives values of C_Q computed in this manner, while Fig. 7 shows a curve of C_Q plotted against Q.

19 *Relation between "Q" and Angle "A."* By solving Equation [**6**] for H_A and introducing the constant C_Q so that Q may be substituted for Q', we have

$$H_A = \frac{Q^2}{C_Q^2 D^2 (8.02)^2 (H - L_1 \cos A - T \sin A)^2} + \tfrac{1}{2}(H - L_1 \cos A - T \sin A) - \frac{V_1^2}{2g} \quad \text{[7]}$$

By solving first for H_A and then substituting this value in Equation [**5**], the value of the constant may be found. Let $C_M = C_K$ in this case. This gives the relation between Q and A by means of two equations and is of value from a theoretical standpoint in showing that C_K approximates a constant. The expression for H_A in Equation [**7**] might be substituted for H_A in Equation [**5**], from which, with the proper value of C_K, A or Q could be found by trial if either of them were given. This would make possible the predetermination of the Q-A curve for any design of gate, provided C_K is properly assumed. In actual operation Q would be read directly from a Q-A curve such as in Fig. 8.

20 *Sample Computation of "C_K."* Using the same data of setting No. 4 as for the computation of C_Q, given in Par. 18, assume an average value of $C_Q = 0.64$ as read from Fig. 7. $V_1^2/2g$ must be found by trial. In this case, however, the actual value of H_A taken from Fig. 6 may be used to compute it, giving $V_1^2/2g = 0.0303$ ft. as computed above. Substituting the numerical data in Equation [7] we find $H_A = 0.585$ ft. Substitute this value for that of 0.591 as given for H_A in the data for the computation of C_M as in Par. 17, and proceed as in the computation of

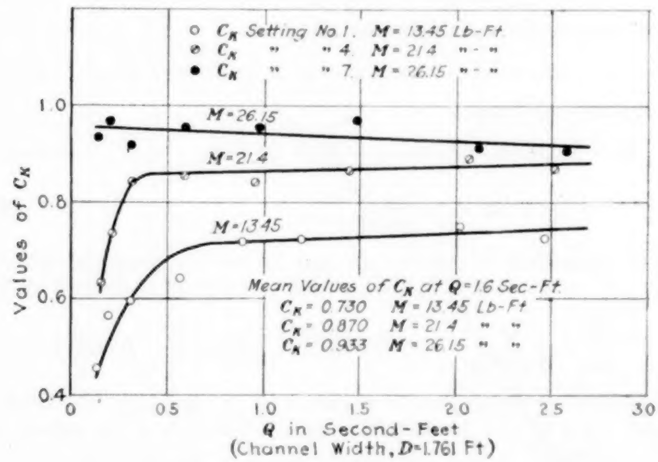


FIG. 9 VARIATION OF C_K WITH Q

C_K, obtaining $C_K = 21.4 \times 0.500/12.41 = 0.869$. Table 2 gives values of C_K thus computed, and Fig. 9 gives a plot of C_K against Q.

DISCUSSION OF CURVES AND RESULTS

21 *Accuracy of Adjustment and "Q-A" Rating Curves.* The points plotted for the second adjustment of the gate, Fig. 8, fall on the curve drawn through the points plotted for the first adjustment. The average deviation of the plotted points from the curve is about 1.5 per cent, and the maximum deviation is 6.1 per cent. This curve would indicate that the adjustment of a gate with respect to bottom clearance, alignment, quadrant centering, and zero adjustment can be duplicated within at least one per cent, and that a rating curve should be correct to within from 0.5 to 1.5 per cent for a given setting.

22 The *Q-A* rating curve becomes approximately a straight line on log paper. Over the working range of from 0.5 to 2.8 sec-ft. the curve is a straight line on log paper, with a slope of 14.45/6.38 = 2.26 for setting No. 4. (See Fig. 10.)

23 *Effect of Changing Static Moment "M."* Fig. 8 shows rating curves for different values of M. For the same value of A, as M increases Q becomes larger. Other factors being constant, the greater the value of M the greater the back-wave clearance, and vice versa. For standardization it is important that the back wave always remains about 0.2 ft. below the gate (see Fig. 5), since the relation of Q to A changes when the wave reaches the gate. All data were taken with the back wave clearing the gate at least 0.15 ft.

24 Values of C_K for different values of M are plotted in Fig. 11, which shows C_K to be practically a straight-line function of M.

TABLE 3 LOSS OF HEAD IN THE INTEGRATING GATE

Q	C	H_A	$H_A - C$	$Q/1.77$	H_w	$\frac{H_A - C}{H_w}$	F	$\frac{H_A - C}{F}$
0.5	0.254	0.398	0.144	0.282	0.173	0.832	...	...
1.0	0.375	0.520	0.145	0.565	0.271	0.535	0.18	0.801
2.0	0.551	0.685	0.134	1.130	0.424	0.316	0.33	0.406
2.8	0.665	0.780	0.115	1.58	0.571	0.202	0.50	0.230

The values of this curve were determined from Fig. 9 for $Q = 1.6$ sec-ft., and are considered average values over the working range.

25 *Individuality of Gates.* To insure the possibility of duplicating the Q-A rating curve with another design of gate, gate No. 3 was designed to have the same static moment as gate No. 2, although it actually was 4.35 per cent greater than gate No. 2. Fig. 8 shows the rating curves of these two gates determined in settings 4 and 6. The maximum difference between these rating curves over the working range is about two per cent. Hence it is possible to duplicate the rating curve with a different gate within practical limits of error.

26 *Individuality of Positions.* Rating curves for settings 3 and 4 are given in Fig. 10. The maximum difference between these curves is at about 0.5 sec-ft. and amounts to 4.5 per cent, dropping to about 2.5 per cent at 2.7 sec-ft., and giving an average difference of about 3.5 per cent over the working range. A portion of this discrepancy can be explained. About half of it is due to increased clearance under the gate, and some to difference in channel width and to differences in gate adjustment. The total error due to individuality of setting is about one per cent. It may be safely assumed that with careful construction, the individuality of setting will not exceed two per cent.

27 The downstream conditions were practically the same in each position, since the back wave always remained at least 0.15 ft. from the lower edge of the gate, which was determined to be the essential condition. Changing the velocity below the gate made no difference so long as the back wave maintained this minimum distance from the lower edge of the gate.

28 *Effect of Channel Slope Upstream.* The maximum difference

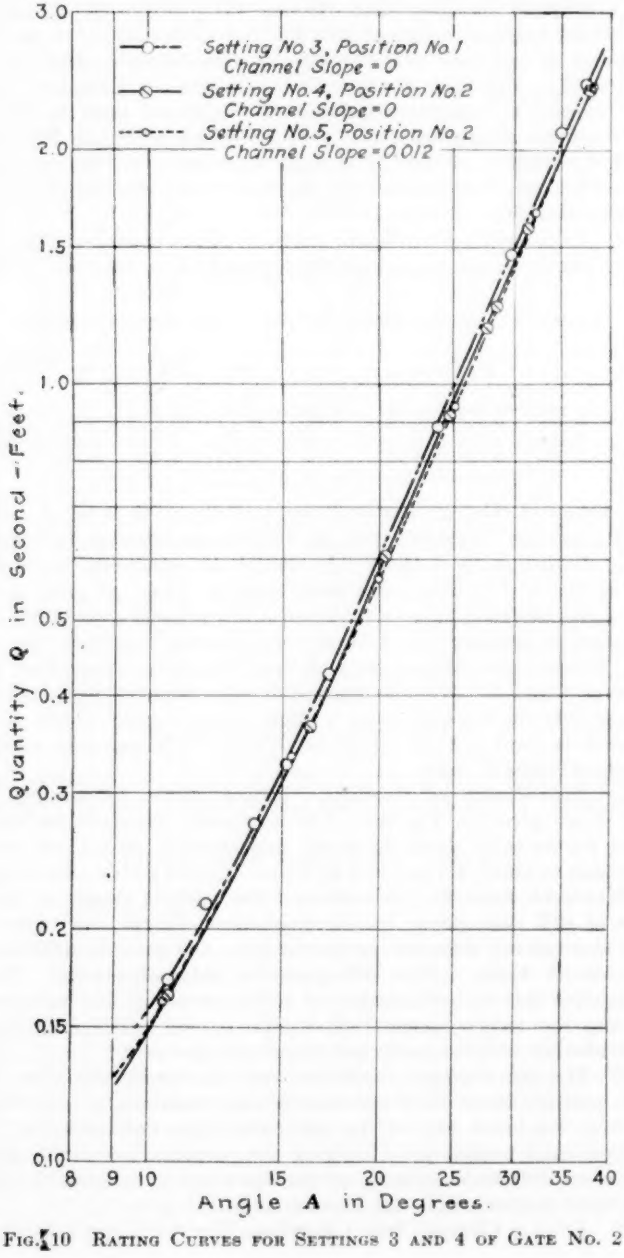


FIG. 10 RATING CURVES FOR SETTINGS 3 AND 4 OF GATE No. 2

in rating curves with the channel level and with a slope of 0.012 was about 1.5 per cent. Hence this device is so insensitive to variations in channel slope that it may be used over a considerable range of slope without error.

29 *Effect of Baffle Board.* The baffle board was uniformly placed

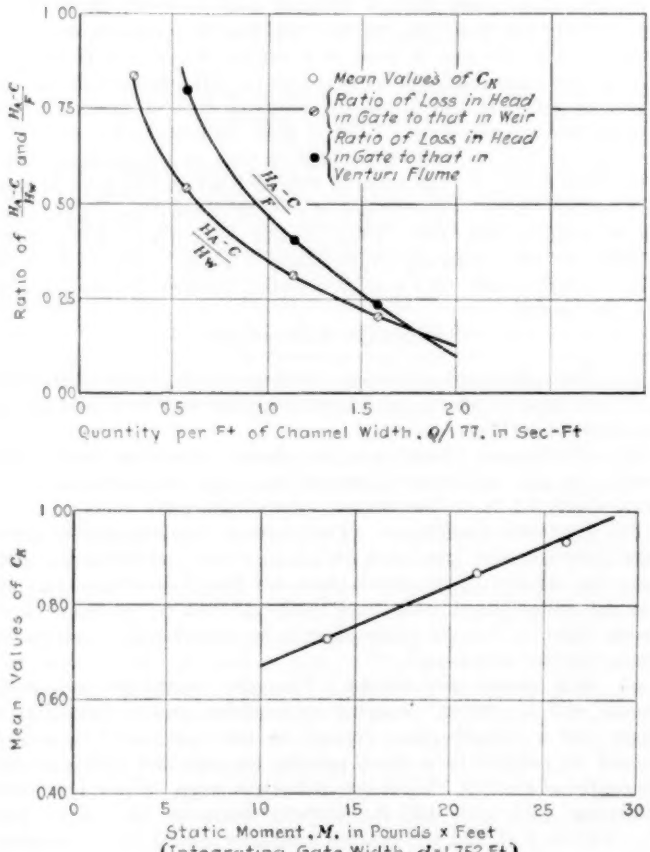


FIG. 11 VARIATION OF C_K WITH M

1.5 ft. upstream from the gate axis and extending below the surface of the water about ½ in. At $Q = 2.4$ sec-ft., removing the baffle board increased A about 7 min. of arc. As A decreased, the effect of the baffle board decreased. The tendency of the pointer to vibrate was reduced about 50 per cent with the baffle board in place, increasing the accuracy of the readings.

30 *Loss of Head.* The loss of head for the integrating gate is $(H_A - C)$, where H_A and C are the depths of water above and below the gate, respectively. C is measured below the back wave. H_w is the head required for a suppressed weir discharging the same quantity per foot of crest as the gate per foot of channel width. This was determined from a log diagram of discharge per foot of crest given on page 213 of Hughes and Safford's Hydraulics, from which the head due to the velocity of approach was subtracted. F is the loss of head in a venturi flume of 1 ft. throat, having the same discharge per foot of its 3-ft. channel as the integrating gate. $Q/1.77$ is the quantity per foot of channel width flowing through the gate. Table 3 gives values of $(H_A - C)/H_w$, the ratio of loss of head in the gate to that in a suppressed weir, and of $(H_A - C)/F$, the ratio of loss of head in the gate to that in a venturi flume. The values of F were taken from Fig. 4, page 18 of Bulletin 265, Feb., 1921, of the Agricultural Experiment Station of the Colorado Agricultural College. Fig. 11 shows $(H_A - C)/H_w$ and $(H_A - C)/F$ plotted against Q per foot of channel width.

Standard Conditions

31 For agreement of results, settings should be standardized. The following standard conditions are suggested from a study of the various settings.

32 *Downstream Conditions.* A velocity should be maintained under the gate sufficient to prevent the back wave coming closer than about 0.2 ft. to the bottom edge of the gate.

33 *Upstream Conditions.* The channel floor should be maintained smooth and level at least 2.5 ft. above and below the gate axis, by using a $\frac{3}{32}$-in. metal plate for the floor of the channel. Use no baffle board. Channel intake should be at least 15 ft. above gate. Channel slope should be determined, and rating curve for that slope used.

34 *Gate Design and Setting.* The gate should be made sufficiently stiff to prevent warping or buckling, should have square edges and a smooth plane surface on the upstream face, which should be parallel to a plane passing through the gate axis and its center of gravity. Thickness at bottom edge, $(2T) = \frac{1}{4}$ in.; side clearance, $(D - d) = 0.01$ ft.; bottom clearance, $(H - L_1) = 0.01$ ft. Values of H and the bottom clearance should be measured to the nearest 0.0005 ft. Adjustment of H and the bottom clearance should be provided by a slow-motion worm under the bolts holding the pivot pins, which should be of tool steel. The axis should be level and perpendicular to the center line of the channel. Standard values for H and d should be adopted. The correct values of these dimensions for gates of different capacities can be determined only by further experiment.

No. 1900

THE ENGINEER: HIS ABILITIES AND HIS PUBLIC OBLIGATIONS

By John Lyle Harrington, Kansas City, Mo.
President of the Society

ENGINEERS have existed in all ages and civilizations as their works abundantly disclose, but until very recent times engineering was an art rather than a science, and engineers were considered superior artizans, rarely deemed worthy of note by the writers of history. They belonged to a class inferior to the warrior king and his nobles, the priest, the lawyer and the physician, all of whom figure materially in the records. In considerable measure engineers are still looked upon and still regard themselves as glorified mechanics. Many still rise by self-education from the ranks of the artizans to high place in the profession. In a recent address to The Institution of Civil Engineers one of its eminent members said: "Engineers . . . have, most of them, owing to the very wise system of their training, had experience in manual work, have worked side by side with men who are now merely wage earners." Though for more than fifty years engineers in increasing numbers have been graduated by schools of the highest standing, though their scholastic training is quite as long and severe as that required for the other learned professions, something of the old view of the profession still remains in the public mind. It requires centuries to change a commonly held view, to break down long-established prejudices, but sometimes a cataclysm hastens the process. The great war was such a cataclysm, for it brought home to the public in spectacular manner the fact that modern warfare rests squarely upon industry, and both upon engineering; therefore it materially raised the profession in public esteem.

Yet there is something lacking. Engineers find it necessary to insist upon the fundamental character of their profession, to point out and to discuss its essential qualities, to call attention to the sound training required in preparation for it; then, to deplore its lack of recognition, its inadequate rewards in money, place, and honor, and to argue the means of obtaining them.

Presidential Address at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

They resent the views so often expressed or implied that the engineer is hardly a scientist, is at most a technician, useful in a narrow sphere but of small value in the important fields of business, finance, and government. In the public mind he is still commonly pictured in khaki, high boots, and a broad-brimmed hat, bossing the job, a worker rather than a thinker. He has secured a foothold in the management of engineering enterprises, but grudgingly and not so generally as he believes to be his due. And while there are abundant examples, cited with pride, of engineers at the head of railroads and constructing and manufacturing enterprises, the number is still comparatively small. In general, the public does not feel certain of the engineer in such positions, does not quite expect to find him there. The financier rarely relies upon him in business, except as an officer of a corporation, where he is guided and supported by business men. It is believed that his knowledge of business and finance is inadequate for safety, that he is something of an optimist and a dreamer, unable to see the pitfalls in a project, unable to evaluate the business element of it, and therefore an unsafe guide for the investor of money. In government the engineer is entrusted with technical matters only, leaving business matters to business men and the dominating political matters to politicians, who hear and sometimes heed the voice of the people, who consider the requirements and determine what is best from the broad, political point of view, untrammeled by the idealism, directness, and honesty of the engineer.

The Need of the Engineer in Government

Our country is still governed by the military and legal professions, influenced somewhat by the business man. Every president, except Harding, has been a general or a lawyer, while the latter not only occupies his peculiar field, the judiciary, but dominates in the legislatures and in the executive branches of the Government. These facts have materially retarded the progress of political development, for the lawyer lives by precedent. His eyes are always upon the past. It results that we progress slowly, that order and rule are deemed more important than service and progress. The engineer with his constructive, creative, scientific mind is sorely needed. His preparation should be broadened so that he may deserve and receive the same public confidence in his human qualities and in his understanding of business and economic and political matters that is now given to his honesty of purpose and to his scientific knowledge. His ideals should be turned more away from himself and his immediate interests and should include appreciation of his obligations to give liberally of his talents, his time, and his energies to the broad service of his community and his country. He should come to understand that by coöperating unselfishly with

his fellows, individually and through his technical societies, he is doing his most to advance the interests of his profession and of his country; that he is developing the high professional consciousness which will give his profession cohesion, weight, and influence in proportion to its potential deserts.

Shortcomings of the Present Educational System

Engineers owe their profession and the country thorough consideration of the essentials of that education which will bring engineers to their fullest development; to education in the collateral subjects so essential to their broadest usefulness. They are fully aware that the future position of the engineer will be determined largely by the plan of engineering education; that the present technical education does not meet all requirements; and the investigation into the matter now about to be made should solve the problem and serve as an example for other investigations of far-reaching character.

But engineers cannot stop there; since they play an increasingly important part in industry, since the welfare of the workers is largely in their keeping, it is their obligation to determine and to bring into use the type of education for workers most conducive to their usefulness to society and to themselves, and consequently to their greatest happiness and highest development. Our education was initially planned to be cultural only; the individual secured his vocational training by apprenticeship or by chance, and we continue, with trivial exception, to train all youth culturally but to leave to fortune training for their economic place in life. We glorify mental labor and train away from work with the hands till Americans consider it degrading, socially inferior, and suitable only for the ignorant, the immigrant, or those of low mentality. The trade unions restrict apprenticeships and the schools offer little education in the trades. Apparently it is expected that the native American, with his superior cultural education, will make his way by his wits without soiling his hands. He is barred by his false social ideals and by his lack of training from those profitable and honorable fields of industry which involve manual labor. Girls are so taught that they look down upon the man who works with his hands. Engineers know these conditions and know their detrimental effect upon men and upon the economic welfare of the country. It is their obligation to present the truth and to press for the reform of our educational system to conform to it. They are not, in general, educators, but they know the need for educating men for industry and they should lead in the endeavor to secure it. The employer is not to be blamed if he demands that the door be opened to foreign labor when he finds it impossible to satisfy his requirements at home. We have seen the building trades throttled, our housing needs un-

satisfied, by the extravagant labor costs resulting from restrictions in training and apprenticeship so severe that these crafts will almost die out with the present generation of workmen.

Engineers are applied scientists, but they have been leaving to private interest or individual initiative the research work in their field. In very recent years they have set up some agencies which have begun to function, but their efforts have not always been so successful that the ability of engineers in this line may be considered brilliant; neither have they demonstrated their ability to coöperate with other research organizations not strictly engineering in character. It is all very well to assume that the engineer fully understands just how engineering research should be conducted, that he can do it better than any one else, but the correctness of these views is yet to be determined. The evidences are not all in the engineer's favor.

The Engineer's Aid Needed by the Economist

The engineer has created modern industry, has been responsible for a production of goods that in this country is substantially sufficient, if other factors were in accord, to establish the universal comfort and well-being that the race has sought since the beginning. But the engineer has been so intent upon devising means for production, upon improving quality of product, and upon managing the men and tools of production to secure the maximum results that he has failed to develop his ability and to establish his position in the other leading factors of life, to which he is entitled by virtue of his part in production and the resulting welfare.

One of the greatest causes of unrest on the part of the worker lies in his lack of understanding of the value of the other factors necessary between the basic raw materials and the consumer of goods. The engineer, in his present position of impartial technician, ought to be the agency to determine what should be the difference between the cost of production and the cost to the consumer, of exposing the truth and righting the errors and abuses now existing, then of showing labor what is its just portion.

The whole business of the country may be likened to a sea on which there are extremely high and low tides at rare intervals; regular and moderate tides at frequent, definite intervals; long ground swells coming from distant disturbances; small waves often violent and destructive, resulting from local causes; but all clouded in mystery to any but the seawise. Capital and men flow from the less to the more profitable branches of industry until the rush coming in from all sides brings overproduction, stagnation, loss, and finally readjustment to the studied and understood needs. These shifts go on in every industry, consequently the balance is never complete, and some

industries are very prosperous while others are greatly depressed and the remainder range between. Business men have little faith in the engineer in business; but his scientific methods applied to those problems of business which are too complicated for the business man's solution by judgment, should result in better understanding of the fluctuations in business and their causes, and finally in remedies which would greatly alleviate, if not cure, the fluctuations which cause panics and unemployment. At this moment the most completely industrialized nation, with all its machinery of production intact, highly organized, amply financed, at peace, is unable to adjust itself to existing conditions; two millions of its workers are idle; more than a million and their dependents are scantly sustaining life on a dole taken by taxes from capital and the earnings of the employed. This nation will muddle through, but scientific principles applied with broad consideration of all factors should prevent such a disastrous breakdown in industry. Countries, like men, sometimes find themselves unable to cope with unforeseen factors, but generally the causes of depression are apparent after the fact, whereas scientific study should disclose them in advance and permit preventive rather than remedial measures to be applied. The engineer should turn his attention to these matters and aid economists to solve the vital problems.

THE ENGINEER AND TRANSPORTATION PROBLEMS

The engineer has already established the fact that he and his methods can secure efficiency in industry and great reduction in the cost of production, and, since the railroads are no longer the football of speculating financiers, the engineer's services are producing beneficial results in transportation. Much remains to be done in the means of transporting goods by water and rail, while the newly developed business of trucking and the hardly touched business of combining the truck with rail and water transportation has yet to be worked out. The combination of means and the proper adjustment of them to the need for low cost or for quick delivery or for both have still to be established. Means which should be supplemental are now more or less destructively opposed, and the engineer is sorely needed to bring about the economies of harmony.

Water transportation, especially on our inland waterways, is ineffective, practically non-existent, because sentiment and commercial-club types of effort have prevailed instead of sound economic and egineering methods. The problem of inland water transportation lies largely in the handling of materials to and from vessels, purely an engineering problem, whereas public attention has commonly been directed toward depths of channel and establishment of boat lines to the neglect of terminal facilities.

The cost of doing business, that is, the taking of goods from the producer and delivering them to the consumer, is in many industries greatly out of proportion to the service rendered. The orchardist who purchases and prepares the land, plants, cultivates, trims, and cares for his trees for seven years before they come into production, then sprays, picks, packs, and markets his fruit, is thoroughly convinced that the business of distribution is unwisely handled when he sees the consumer pay four to five times as much as he receives for his product. The apricot growers of California knew somebody blundered when they were paid $150 per ton for their fruit one year while the next they were without a market because the canners did not sell the previous year's pack. In all such industries the entire risk is thrown upon the producer because he is less well organized than the merchant, but the whole problem of marketing needs scientific study.

Scientific Fact Finding the Engineer's Prerogative

The judgments regarding the ability of the market to receive and pay for goods are unscientific and too generally defective. We now have capacity for making automobile tires quite out of proportion to the need for them, of producing coal far in excess of the country's requirements; we are producing wheat in much greater quantity than the markets of the world demand, and at many points industry is out of balance. There is no sound objective to the doctrine of *laissez faire;* it is probably unwise to try to curb the freedom of the initiative of the individual or of the corporation in honest business; but scientific fact finding, the engineer's prerogative, employed before the venture is made, would save both capital and labor from disastrous results. The judgment of the business man could well be supported by the engineer's science.

Sound and scientific adjustment of industry, based upon knowledge of all the facts, would prevent the great diversity of reward among those engaged in fundamental industries. It is difficult for one of the unorganized laborers and tenant farmers of the South, who rarely receives in a year as much as $400 in money and goods and the rent of a poor house for the labor of himself and his family, to believe governmental agencies to which he contributes through taxes are justified in requiring the railroads to pay four times this sum to the average railroad worker of the same locality on the theory that it is essential to proper standards of living. If the Government were scientifically administered all facts would be available, and equity as between industries and localities would be reasonably well secured.

The Engineer and Public Business

Engineers have not until recently disclosed an aptitude for public business and governmental matters. They have followed the habit of employees and devoted themselves to the interests of their individual employers. And with that lack of ability to coöperate with each other for which they have acquired an undesirable reputation, as evidenced by the multitude of their organizations as compared with those of other professions, they have only recently begun to employ their special talents in the public service. The investigation of the twelve-hour day disclosed the truth, and results naturally followed. The investigations and reports made by interested business men or disinterested sentimentalists accomplished nothing, because they were biased and therefore not generally accepted; but the engineers' report carried with it conviction of ability to determine and honstly to state the facts, hence it was promptly accepted and acted upon.

It naturally follows from the engineer's custom of dealing with the unalterable laws of nature that he generally presents the facts as he finds them. It is, of course, not uncommon for the engineer in subordinate positions to color the facts to suit his employer's interest, but it is less common than in business or in other professions. It is the engineer's habit to be fair between client and contractor, between buyer and seller, between employer and employee, between capital and labor; and on account of this custom, born of his scientific methods and habits, he is better fitted than any other to bring capital and labor together on a sound basis and thus to stop the enormous waste their quarrels now cost. Management is by no means free from the mind of the feudal lord; and labor is too generally dominated by leaders who make a living out of making trouble, out of endeavoring to secure for labor more than its just share of the goods it helps to produce and, for labor in his particular industry, more than its just share of the total earnings of labor in all industry. Out of ignorance or cupidity, labor leaders too often distort the facts, while management at times, by secrecy or deceit, secures too great a share for capital. The result is distrust, unfair demands, strikes, lockouts, and loss to both parties and to the public. The facts set forth impartially after the methods of the engineer would greatly reduce if not eliminate these bad practices.

It is not the purpose to argue that the engineer is the one logical man to perform all industrial functions, that he should entirely replace the lawyer as an adviser, that he should take over all the functions of the business man, that he should replace the financier, or that he should take over the functions of government, beyond the extent to which his place in industry and his peculiar fitness make wise; but it is imperative before

we proceed much further that the engineer perform his proper part and office in all these matters.

Engineers in municipal government, which is chiefly engineering and business, are demonstrating their worth. They come to the work with unusual equipment: technical knowledge, honesty of purpose, the creative mind; but the world is reluctant to admit that they have the business and the political qualities essential to the work. It is usual to thank fortune that engineers are not politicians, to deprecate the services of politicians because they are too commonly spent for the politician instead of the public; but though a politician is usually one who is in politics for private advantage, he may be one skilled in political science and administration; and if the engineer be equipped in this latter sense also, he should be an ideal municipal public servant.

In state and federal matters engineers have only recently ventured beyond their technical field. A whole administration of the Federal Government apparently subscribes to or at least acquiesces in the view that an engineer is not equipped to administer an engineering bureau or to conduct business affairs founded on engineering. It is true that this is politics of the questionable kind, but the lack of public confidence in the engineer's business and administrative ability makes possible its success. No one is disturbed about it but the engineer.

The United States developed in a century from an agricultural nation in which each household produced its own food and manufactured nearly all the goods necessary to satisfy its needs to a highly organized industrial nation producing goods of all kinds in quantity and at a price never before known, though wages in terms of money or goods or benefits are not equaled in any other country. The engineer has enabled the industries of this country to compete in the markets of the world with the handicraft nations with their vastly lower standards of living. We are too prone to congratulate ourselves on our progress, to be content as a whole people with the successes we have achieved, forgetting that in the warring of the groups or divisions of industry the individual, sometimes whole groups, frequently suffer; but there is always a material element of society which is too weak to hold its place; and in sacrificing it we are sacrificing the whole well-being of society. It is not a solution of the problem for the strong to halt long enough to give of their production to the weak: that is but a sop to man's sense of responsibility for his brother; the strong are obligated to help the weak to help themselves, to curb the predatory, and to bring to a higher level the lowest standards of living.

The engineer is a scientist trained in the application of the scientific method to all his problems, accustomed to determine and to be guided by the truth. It is his part to determine and present for the use and enlightenment of the public the whole

truth about the industry he has created. When business, through greed or ignorance, abuses industry, a clear exposition of the facts is the surest means of setting the matter right. When self-seeking politicians attack industry for their own benefit but to the injury of the industry and the country, it is the obligation of the engineer, individually and through his organizations, to present the facts which will set the public mind straight. The energy to determine the truth and the courage to present it will go far toward curing the evil.

The Engineer Politically Weak

In some cities we see organized engineers consulted and heeded in civic matters; in others we see them utterly ignored in the expenditure of public funds in works with which the engineer is particularly qualified to deal. Politically the engineer is weak because he rather expects his opinion regarding engineering matters to be accepted by the public and by politicians simply because his professional knowledge enables him to speak with assurance. He fails to appreciate that it is not enough to know, to be right, but that he must present his views with vigor and true political wisdom if they are to be adopted.

Because the engineer rarely presses his view on the public or upon the business organizations with which he is associated, he likes to think he is modest. He likes to sit in the background till deference is shown to his superior technical knowledge by calling him into conference. Too often the reticence he considers modesty springs from lack of confidence in his ability to present his views convincingly; for he commonly knows that he is weak in forensic and political ability. Is it not possible that his modesty is really pride in his technical knowledge and fear that he is unable to enforce his views?

The Engineering Profession Ineffectively Organized

The engineer considers himself competent to organize certainly industry, possibly business and government, but he has not demonstrated the ability to organize most effectively his own professions. Instead of one comprehensive organization designed to deal with the social, the welfare, the professional, the research, the standardizing, and the public-service matters with which engineers are concerned, groups have formed many societies busied principally with professional and social matters. Intolerance or lack of foresight or inability to get on with their fellows caused each group to seek opportunity for self-expression by forming a separate organization devoted to the particular, narrow interests of the group. The splitting-off process continues and will continue unless the engineers awake to the harm it is doing. Engineers, more than any other group, find diffi-

culty in composing their differences, in harmonizing their efforts. The four great founder societies are learning to work together, somewhat as the result of the efforts of the great iron master, but harmony has come slowly, and not without jealousy and suspicion. The lesser groups are more or less ignored and some of the newer ones are materially antagonistic to the older organizations. The differentiation should never have taken place; the profession should have developed within one all-inclusive unit, which, by numbers, by its ability to speak for all engineers, as the Bar Association speaks for all lawyers, could exert the whole weight of the profession in industry and in public affairs.

The profession is not yet entirely differentiated from the trades from which it arose. It is true that within fifty years great numbers have been graduated from our technical schools and been received into our engineering societies with that guarantee of their training, but so many competent men are still self-educated that our engineering societies provide for their admission to membership. The only assurance obtained regarding the character of their professional training and regarding the fitness of all candidates for the higher grades of membership, is the possible view of five members of the society, whose opinions may be more generous than sound. We have need for every grade of engineer and rightly have appropriate grades of memberships in our principal engineering societies. The British societies receive proportionately fewer university-trained men, hence they assure themselves of the candidate's fitness by requiring that he pass an examination into the sufficiency of his professional knowledge to warrant giving him the professional status membership confers. Engineering societies in this country might well pursue a like course, so that membership in any grade will establish the professional status with a higher degree of assurance.

Licensing and the Status of the Engineering Profession

Opinion is greatly divided regarding the advisibility of making engineering a closed profession, or fixing by legal license the professional status of every private practitioner and every engineer employed by others. The leading proponents of license laws advocate such meager requirements that the license may be had by any one calling himself an engineer. Many of the states which require engineers to be licensed make the tyro and the expert equal before the law, make the license meaningless and add to the present confusion in the minds of laymen. A few states require examinations and records that expose the real engineering status of the licensee. Engineers complain that unqualified men assume the title, and often, through ignorance of the employer or design of public officials, attain the place of

qualified men to the distinct harm of the profession. It is established that the only way to prevent the unqualified from assuming the name and claiming competence is by closing the profession, as has been done in law and medicine; but it is argued, and with reason, that the creative mind is not an academic matter; that this will bar from service many good men whose natural ability and whose training in certain narrow lines fit them well for limited service. It is further argued that it is not practicable for a license to show the specific fields within which the licensee is entitled to practice. That is true in law and medicine, but no great difficulty flows from it in those professions. If licenses are graded, much as the members of the founder societies are graded, if they show the principal divisions of engineering in which the licensee is authorized to practice, if they are granted only after suitable examination and are uniform in requirement and reciprocally accepted among the states, the license will improve the status of the profession without placing undue hardship upon its members.

The profession must cleanse itself of bad practices. The fundamental principles of its ethics are well understood and clearly enunciated, but our codes offer limited control because they are administered, if at all, by committees of our societies who are loath to take on the functions of judge and executioner. Cases of infraction are rarely clear and unequivocal, and some kind of hearing or trial is essential to establishment of the facts. The injured party will rarely subject himself to the criticisms, annoyance, and loss of time this entails. It results that he complains privately but takes no steps to secure justice, and the violator of the code is encouraged to repeat the offense. If the trial body were better established and clearly performing a public duty, as would be the case if a licensing board formed the tribunal, the situation would be greatly improved, in so far as the flagrant abuses are concerned.

The profession would, however, still have the difficult duty of purging itself of petty jealousies, of unfair or otherwise improper criticism. It is impossible entirely to suppress these evidences of the weakness of human kind, but by constant endeavor they can be greatly reduced, as they are in the older professions. Thorough attention in the schools to instruction in ethical standards, constant reiteration of them in the professional societies, will increase their observance and enhance the character and position of the profession.

Engineers Must Address Themselves to All the Broader Problems of Industry

To gain the position in the business and financial fields to which the profession considers itself entitled, engineers must cease to deal so exclusively with technical problems and address

themselves to all the broader problems of industry. The line between technical and business matters cannot be sharply drawn. The engineer has entered the field of management, and the rewards and the respect for his abilities are appearing. He must go further, and deal with the problems in sales, markets, and the financial structure of the industries in which he is engaged, and become a partner rather than a subordinate in the business.

But in the broad study of industry as a whole as it affects the nation and the world the engineer should have an important part. He should contribute largely to the solution of the problems of the equalization of industry; of preventing inflations and depressions, for they are complementary; of encouraging development in lines in which competent investigation shows it will be rewarded; of retarding those lines which show a tendency to excessive development. This can best be done by a thorough analysis of the factors entering into the problem and publishing the results to the world. The profession should resolve itself into a fact-finding and publishing organization in all matters pertaining to the industry with which it is associated.

Along with this, the profession through its individual members and through the coöperative efforts of its organizations must deal with politics, for government has great influence upon business and industry. The engineer, individually, must do his full political duty, must bear the onus of standing for office, must enter the political contest as the members of other professions do, and must give freely of his time and special knowledge in the public interest. He must forget his modesty and fight for place and authority if he is to receive them and render service.

And when the importance of the matter to industry or to the public welfare warrants, the whole weight of the organized profession should be employed to secure wise administrative or legislative action. Engineers have as much obligation as any other group to urge the adoption of their views in matters of their particular knowledge and interest. The profession must seek to serve the nation thoroughly, unselfishly; must give freely of its means and of the time, energy, and knowledge of its leaders, who must be supported by the engineering organizations acting together for the common good. The profession must not wait to be invited into the councils of nations, but must make its opportunities and its place. Public service should be rendered with reasonable modesty, leaving to the fairness of the public mind recognition and appreciation of it. To do this it is essential that engineering organizations put aside pride and prejudice and bury all differences in service. They may preserve their individuality in all matters of their particular interest, yet join without reservation in some acceptable plan in all matters of common concern, forgetting everything but that there is service

to be rendered and that the entire weight and talent of the profession can perform it to the best advantage of all.

Engineering is a young profession and is still in a state of flux. It is not bound down to tradition nor to precedent, but has the virility of youth; the courage, the energy, and the orderly and creative mind essential to the solution of not only its own problems, but of all the great problems of the industrial world. Those problems are both technical and economic, and upon their satisfactory solution the prosperity and the peace of the world and the progress of the race depends. The responsibility is great, but we proceed with confidence that the profession will ably meet its obligations.

No. 1901

ANNUAL REPORT OF THE COUNCIL 1922–1923

THE Council presents herewith its report to the membership covering the presidential term of John Lyle Harrington, and containing the reports required by the Constitution upon the status of the Society.

COUNCIL MEETINGS

Six meetings of the Council have been held and six meetings of the Executive Committee in the intervals between Council meetings.

COMMITTEES OF COUNCIL

Reports of the Standing Committees of the Council are made a part of the Council records for the year and cover the detailed activities of these committees which have also been reported from month to month in *Mechanical Engineering* and the *A.S.M.E. News*. The committees are listed on pages viii-xii of this volume.

FINANCES

The audit of the books for the fiscal year and the report of the certified public accountants is made an appendix to this report.

In connection with the study of the financial situation and the preparation of the Budget for the year 1923–1924 is an investigation directed by the Council under a special Policy Committee, Dexter S. Kimball, Chairman. This committee's report was published in the *A.S.M.E. News*, November 7, 1923, and resulted in the Council's presenting an amendment to the Constitution to increase the dues. At the Annual Business Meeting of the Society this amendment was ordered to be balloted upon by the membership.

MEETINGS OF THE SOCIETY

The Spring Meeting was held in Montreal, Canada, May 28–31, 1923, the Engineering Institute of Canada coöperating. An attendance of 631 members and guests was registered and 10 technical sessions were held.

The Annual Meeting was held in New York, December 3–7, 1923, with a registration of 1852 members and guests; 18 technical sessions and over 30 committee meetings were held.

The Regional Meetings held in Chattanooga and Los Angeles are reported under Local Sections, the Meetings and Program Committee having jurisdiction over these meetings with the Local Sections.

PUBLICATIONS

The Society has published 12 monthly issues of *Mechanical Engineering;* 24 semi-monthly issues of *A.S.M.E. News*, and Volumes 43 and 44 of the Transactions, containing 1334 pages and 1421 pages, respectively.

In order to complete the Society's program of publications without exceeding financial limitations the Society furnished TRANSACTIONS in paper

bindings. The morocco binding was supplied at cost to such members as desired it. The Year Book was not issued in 1923. The Engineering Index Annual for 1922 includes over 16,500 items. Condensed Catalogues, the thirteenth annual edition, shows a gain over previous issues of 393 firms and includes the Mechanical Equipment Directory having over 30,000 listings.

MEMBERSHIP

There have passed through the hands of the Membership Committee during the calendar year 1923, 2569 applications for membership or change of grade, of which 2312 have been approved by the Council, as follows:

Members	499
Promotions to Member	157
Associates	65
Promotions to Associate	0
Associate-Members	557
Promotions to Associate-Member	114
Juniors	920
Total	2,312

Student Associates. Provision has been made for waiving dues of Student Associates for one year after graduation, the initiation fee only being required. This new rule was issued by the Council after careful study and recommendation by the Committee on Relations with Colleges and the conference of Student-Branch delegates. The rule was adopted with the expectation that desirable members would join the Society from one to five years earlier than would otherwise be the case, and that the young engineer would be enabled to secure the influence of membership at a period when he needs it most.

Table 1 on page 274 shows the status of membership covering the fiscal year October 1, 1922, through September 30, 1923.

Memorial notes of members who have died during the year appear in the Transactions. Among the members lost by death is included our Past-President and Honorary Member, Robert Woolston Hunt.

PROFESSIONAL DIVISIONS

In their provision for the technical programs, both for local and general meetings, the Professional Divisions have been very active.

Coöperation of the Ordnance Division with the War Department in the "Preparedness Program" is of special note. This program, by formal act of Congress, authorizes the preparation of adequate plans for mobilizing industry for war emergency; the Assistant Secretary of War is made responsible for the development of a suitable industrial organization.

The Management Division held a second successful Management Week in October in coöperation with the Local Sections and with other management organizations. The Materials Handling Division produced a new formula for economics in materials-handling equipment.

LOCAL SECTIONS

The Local Sections, now numbering 64 geographical groups, have been sponsors for two regional meetings, one at Los Angeles, Cal., April 16–18, and one at Chattanooga, Tenn., October 23–24.

The second Annual Management Week, beginning October 25, was conducted under the auspices of the Local Sections, the Management Division, and several organizations coöperating. The main topic was Methods of Measuring Management. About thirty management meetings were held throughout the country during this week.

CONSTITUTION AND BY-LAWS

The revised Constitution and the Code of Ethics adopted by letter ballot of the membership, are made an appendix to this report.

The adoption of the new Constitution very fittingly came in the term of President Harrington, who, as Chairman of the Committee on Constitution and By-Laws the previous year, directed the important work of preparation of the new document.

The Committee, at the request of the Council, prepared and reviewed By-Laws and Rules covering details of the constitutional requirements. These all appear in the 1924 Year Book of the Society.

AWARDS AND PRIZES

The annual awards this year are:

A.S.M.E. Medal for 1922 to Frederick A. Halsey, of New York, for his paper on the Premium System of Wage Payments, presented before the Society at its Providence Meeting in 1891, "which had a most profound effect toward harmonizing the relations of worker and employer."

A.S.M.E. Medal for 1923 to John R. Freeman, of Providence, R. I. Past-President of the Society, "in recognition of the eminent service he has rendered in fire-prevention work."

Junior Cash Prize to S. S. Sanford, of Detroit, Mich., as principal author of the paper on Elasticity of Pipe Threads, prepared jointly with Sabin Crocker.

First Student Prize to Charles F. Olmstead, of Minneapolis, Minn., for his paper on Oil Burning for Domestic Heating.

Second Student Prize to H. E. Doolittle, of San Diego, Cal., for his paper on The Integrating Gate.

Life Membership to Prof. John Airey, of Ann Arbor, Mich., the principal author of the paper, appearing in Volume 43 of the Transactions, entitled On the Art of Milling; C. J. Oxford was co-author with Professor Airey.

STUDENT ACTIVITIES

Affiliated Student Branches are now conducted by 77 colleges. During the year these Student Branches have held approximately 350 meetings with about 4000 students enrolled; eight new Student Branches have been added. The holding of joint meetings with the Local Sections has been of unusual value in the development of Student Branches in such localities. Prominent members of the Society have been the speakers at these gatherings and have been most generous in their support.

LIBRARY

The Library Board reports the number of registered users as about 34,000, of whom 26,000 visited the reading room. Searches and translations were made for over 300 persons and photoprints for approximately 2500, covering over 22,000 photoprints. Information by mail was sent to some 5000 persons and by telephone to over 3000. The permanent catalog now contains 175,000 cards, indexing the material under some 25,000 subjects. Over 80,000 volumes are now fully catalogued and the recataloguing of the older works is being carried on very actively. The entire work of recataloguing, on which the Library has been greatly helped by the Carnegie Corporation, is now seven-eighths completed.

POWER TEST AND SAFETY CODES, STANDARDS, AND RESEARCH

Power Test Codes. The outstanding accomplishments during the year are (1) the publication of the Test Code for Hydraulic Power Plants and

TABLE 1 STATUS OF MEMBERSHIP — 1922–1923

			Losses				Additions		Totals		
	Oct. 1, 1922	Oct. 1, 1923	Transferred from	Resigned	Dropped	Died	Transferred to	Elected	Loss	Gain	Total
Honorary Members	19	20	...	...	...	..	...	1	...	1	1
Life Members	77	75	...	...	...	4	...	2	4	2	−2
Members	7733	8139	...	43	52	58	126	433	153	559	406
Associate-Members	4023	4414	84	23	21	6	90	435	134	525	391
Associates	844	873	8	10	14	2	2	61	34	63	29
Juniors (15)	680	932	126	19	23	1	421	...	169	421	252
Juniors (10)	3265	3485	421	32	39	3	...	715	495	715	220
Affiliates	55	....	...	...	...	..	...	...	...	...	...
TOTALS	16696	17938	639	127	149	74	639	1647	989	2286	1297

Their Equipment, Code on General Instructions, and Test Code for Reciprocating Steam Engines in pamphlet form; (2) the printing of the preliminary drafts of the Code on Definitions and Values, Test Codes for Stationary Steam Boilers, Reciprocating Displacement Pumps, Displacement Compressors and Blowers, Feedwater Heaters, Locomotives, Internal-Combustion Engines; and (3) the establishment of contact with the recently organized Joint Committee in Great Britain representing eight of the principal British institutions of civil, mechanical, electrical, marine, and chemical engineers.

Standardization. Engineering and industrial standardization of direct interest to the Society has made real progress. Eighteen committees have been at work, and have completed substantial parts of the projects assigned to them under the procedure of the American Engineering Standards Committee, covering Standardization of Transmission Chains, Shafting, Ball Bearings (the latter being developed with the Society of Automotive Engineers and the national standardization bodies of Great Britain, Sweden and Germany); Standardization and Unification of Screw Threads, Pipe Flanges and Fittings, Bolt, Nut, and Rivet Proportions, Code for Identification of Piping Systems, and Standardization of Fire-Hose-Coupling Threads. The Society is also sponsor for the Sectional Committees on Standardization of Elevators, and of Small Tools and Machine-Tool Elements.

Research. There are nine research committees whose activities are being brought more into contact with the Engineering Foundation and the National Research Council and the work of other societies in these fields. A very satisfactory arrangement has also been worked out with the Professional Divisions. The Council's appointee to the National Research Council is the Chairman of our Standing Research Committee.

Bearing Metals Research has been assisted by the Engineering Foundation; work on the Extension of the Steam Table, financed by special subscriptions, is being carried on at Harvard University,

Massachusetts Institute of Technology, and the Bureau of Standards, and an understanding with officers of the Institution of Mechanical Engineers in connection with this research has been initiated.

Notable progress has been made in work on Lubrication, Fluid Meters, Gears, Cutting and Forming of Metals, the latter research specially urged by the Machine Shop Practice Division.

Safety Codes. Activity in this field has been confined principally to the Safety Code for Mechanical Power Transmission Apparatus and the Safety Code for Elevators. The organization meeting of the Safety Code for Compressed Air Machinery was held on May 23, 1923. This committee now consists of 22 members representing 17 organizations. The Society has representation on fourteen other committees and there are in course of preparation safety codes for Abrasive Wheels, Aeronautics, Electric Power Control, Floor Openings, Railings and Toeboards, Ladders, Laundries, Lighting, Industrial Sanitation, Logging and Sawmill Machinery, Machine Tools, Mechanical Refrigeration, Paper and Pulp Mills, and Power Presses.

SERVICE TO THE PUBLIC AND INTERSOCIETY RELATIONS

American Engineering Standards Committee completed its sixth year of service. This Committee was originated in 1918 to coördinate the standardization work of the national engineering societies. It does not initiate projects, but forms and administers policies for the carrying out of the work, while the actual work of securing agreement on standards falls to the sectional committees organized by national societies and associations who accept the sponsorship. Early in the year the committee reported 23 member bodies, 205 participating agencies with duly accredited representatives, and 917 individuals engaged in committee activities.

Boiler Code. The interpretation service of the Boiler Code Committee to the public and the consulting work for the benefit of the Inspection Departments utilizing the rules of the A.S.M.E. Boiler Code, have grown in volume. The Committee has held nine meetings during the year, with several additional meetings of sub-committees. Of major importance has been the revision of the Heating Boiler Section and Power Boiler Section of the Code.

Coöperation Committee of the Founder Societies. The American Society of Civil Engineers is not a member of the F.A.E.S. and coöperation between the four Founder Societies is secured through a newly organized Joint Coöperation Committee. The policy of coöperation followed for many years past by the Founder Societies is to be fostered in the future through this joint committee, consisting of the four Presidents and the four Secretaries who give preliminary consideration to the various problems, suggesting the policy to be followed in each case, and recommending a procedure for carrying out these policies, using as far as practicable existing agencies.

Employment Service Bureau. This service is under the jurisdiction of the four Founder Societies and has been recognized this year, making it a limited free service, available only to members of the four national engineering societies. The policy developed after most painstaking studies by joint committees and a special committee of the Council, will make the employment service partly self-supporting and, it is hoped, more successful.

Engineering Foundation was established in 1914 by Ambrose Swasey, of Cleveland, Honorary Member and Past-President of the Society, "for the furtherance of research in science and in engineering, or for the advancement in any other manner of the profession of engineering and the good of mankind." The Society has two representatives appointed by the Council. Research Narratives have been published and distributed by the Foundation semi-monthly during the year.

Federated American Engineering Societies, now in its third year, has a membership of 29 national, state, and local societies. The A.S.M.E. has 18 members on the American Engineering Council of the Federation.

This organization has many accomplishments to its credit, but with the revision of the constitution and by-laws now under advisement, it will be better equipped to further the public welfare wherever technical knowledge and engineering experience are involved, and in all matters of common concern to the engineering profession.

Some of the problems of national importance are:

Coal Storage, undertaken with the approval of the U. S. Coal Commission and covering a comprehensive treatment of the engineering, chemical, and economic phases of coal storage. There has been a free exchange of information regarding the coal requirements of important industrial districts and groups of industries.

Reforestation: With the aim of directing the attention of engineers in particular and the public in general to the present serious situation due to national and state delays in passing constructive, workable, and non-political laws for afforestation and reforestation.

The Twelve-Hour Shift: A committee having the Hon. Herbert Hoover as Chairman. This report with a personal foreword by the late President Warren Gamaliel Harding and the personal interest he took in the matter resulted in the elimination of the long-shift day from the iron and steel industries of the United States.

The National Board of Jurisdictional Awards has been in existence since 1919 and has arbitrated a great number of jurisdictional strikes, with a resulting decidedly beneficial influence in the building industries.

John Fritz Medal award for 1923 will be made to Ambrose Swasey.

Licensing of Engineers. A special committee appointed performed signal service in assisting the engineers of New York State at a critical time, when it was found that through misunderstandings of the law, many of our members had failed to obtain licenses on May 1, 1923, when the law became mandatory. The Committee was composed of Colonel Junkersfeld, Chairman, and Messrs. A. G. Pratt and John Milton Goodell. A canvass was taken of the members in the state, and a request made to the Governor and the legislature, to postpone the date to August 1, was granted.

The matter is now in the hands of the F.A.E.S. with the hope of securing uniformity of laws in the several states and a supervision that will protect engineers in the practice of their profession.

National Research Council, Division of Engineering, in coöperation with Engineering Foundation, was established in 1910 under the auspices of the National Academy of Sciences to encourage, initiate, organize, and coördinate fundamental and engineering research and to serve as a clearing house for research information in the field of engineering. We have two representatives appointed by the Council, with an additional representative appointed from the Engineering Foundation Board from our Society's representation on that Board.

Society for the Promotion of Engineering Education. John Lyle Harrington and Frank A. Scott were appointed to represent the Society on the Board of Investigation and Coördination, which has obtained funds from the Carnegie Corporation to make a complete survey of engineering education.

World Power Conference. Of international significance has been the invitation extended to the United States to be a participating country in the World Power Conference to be held in London in 1924 in connection with the British Empire Exposition. The several Government agencies, with the technical societies and other special engineering organizations, have organized the American Committee which includes four cabinet officers. The A.S.M.E. has three representatives on the American Committee.

SOCIETY REPRESENTATION

The Society has been represented in the following functions and conferences, the representatives named being appointed as Honorary Vice-Presidents:

Pan-American Commercial Congress, Honolulu, Hawaii: S. N. Castle.

Wellman Memorial Meeting, Cleveland, January: Ambrose Swasey, E. S. Carman.

Ohio State University, Award of the Sullivan Medal: Ambrose Swasey

Annual Dinner of A.I.E.E.: Prof. W. H. Kenerson, Vice-President representing President Harrington.

Annual Dinner of A.I.M.E.: W. S. Finlay, Jr., Vice-President, representing President Harrington.

American Wood Preservers Association: Society coöperating through Forest Products Professional Division, "in steps that will increase the durability of wood products now used and encourage intelligent use of present timber as well as propagation of forests."

Associated General Contractors Association: Coöperation through the Society's Materials Handling Division and coöperation with A.E.S.C. whereby certain standardization work will be assigned to the A.S.M.E. Professional Division.

American Association for the Advancement of Science: Associate, Dr. Alex. C. Humphreys, Past-President, with a special representative each year, depending on the place of meeting of the A.A.A.S. The meeting was held this year in Cincinnati, and Prof. B. E. Muncy was our special representative.

General Committee on Standardization of Ship Construction: Organized by the shipbuilding and marine interests of the U. S. at the suggestion of Hon. Herbert Hoover. Our representatives were J. W. Gray, Philadelphia, and H. H. Brown, New York, alternate.

Seventh-fifth Anniversary of Société des Ing. Civils de France, Paris: Jesse M. Smith, Past-President, and Laurence V. Benet, Paris.

One Hundred and Twenty-second Anniversary of Société d'Encouragement pour l'Industrie Nationale, Paris: Laurence V. Benet.

Twenty-seventh Annual Meeting of American Academy of Political and Social Science: Robert H. Fernald, J. W. Lieb, and Calvin W. Rice.

Fifteenth Annual Conference of National Conference on City Planning: Baltimore, W. W. Varney.

National Air Institute in St. Louis (Second Congress): Prof. E. P. Warner, Chairman of the Aeronautical Division.

Massachusetts Institute of Technology, Inauguration of President Stratton: Ambrose Swasey, Past-President and Honorary Member, and Calvin W. Rice, Secretary.

Institute of Architects, Fifty-sixth Annual Convention and Conferring of Institute's Gold Medal on Henry Bacon, architect of the Lincoln Memorial: A. M. Holcombe.

International Air Congress (London): Prof. E. P. Warner, Chairman of the Aeronautic Division.

American Academy of Political and Social Science, conference on "The Government, The People and the Price of Coal": Fred R. Low, President-elect, H. V. Coes, and James W. Cox, Jr.

Twenty-fifth anniversary of the Society of Mechanical Engineers of Japan, and the Thirtieth Anniversary of the Society of Naval Architects of Japan: Elmer A. Sperry.

American Marine Congress: Spencer Miller and Hosea Webster.

Other annual appointments are the Joseph A. Holmes Safety Association on which General Bixby of Washington has represented the Society for many years; the Washington Award of the Western Society of Engineers, Prof. Herbert S. Philbrick, to fill the vacancy caused by the expiration of the term of Charles F. Brush; and Washington University, St. Louis, Inauguration of Chancellor Herbert Spencer Hadley: E. R. Fish.

CONCLUSION

It has been aptly said that while the function of the Council must necessarily be the transaction of business for the Society, so that the Society may remain the educational agency for our profession, yet always we must endeavor wisely and generously to coöperate with other associations, tending to cement the bonds of brotherhood in the whole profession, and have the ideal of Service to our Country as our ultimate aim.

APPENDIX NO. 1

REPORT OF ACCOUNTANTS

Wm. J. Struss & Co., certified public accountants, give the results of their examination of the books of the Society for the fiscal year ended September 30, 1923, in the following statements of assets and liabilities, and income and expenses. The total of the expenses includes $21,974.42 not paid at the close of the fiscal year.

BALANCE SHEET, SEPTEMBER 30, 1923

Assets

Society's one-quarter interest in the Building, Land, and Real Estate Equipment (25 to 33 West 39th Street)			$486,792.79
Library Books		$ 13,000.00	
Furniture and Fixtures		5,000.00	
			18,000.00
Stores, including plates and finished publications			34,800.39
Engineering Index			10,000.00
Trust Funds Investment			
Liberty Bonds		7,000.00	
Lawyers Mortgage 5-1/2% Bonds		41,000.00	
St. L., Peoria & N. W. 1st 5% 1948 (Par $10,000)		10,613.89	
Cash in banks representing Trust Funds		1,718.63	
			60,332.52
Liquid Assets			
Liberty Bonds		18,000.00	
Accounts Receivable:			
Members' Dues	$ 57,598.61		
Initiation Fees	2,473.76		
Sales of publications, advertising, etc.	114,954.15		
		175,026.52	
Cash in banks for general purposes		7,271.68	
			200,298.20
Advance Payments			52,216.55
			$862,440.45

Liabilities

Trust Funds		
Life Membership	$47,263.05	
Library Development	4,902.71	
Weeks Legacy	1,957.00	
Melville Medal	1,330.50	
Hunt Memorial	254.26	
Hess Junior Prize	1,000.00	
Hess Students Prize	1,000.00	
C. T. Main Award	2,625.00	
Total		$ 60,332.52

Trust funds total, brought forward	$60,332.52
Notes Payable	50,000.00
Accounts Payable	12,795.40
Initiation Fees uncollected	2,473.76
Replacement Fund	1,163.18
Steam Table Research	3,622.01
Dues paid in advance	1,447.73
Unexpended appropriation 1922-1923	21,974.42
Taylor Biography	439.61
Standardization of Bolts, Nuts and Rivets	1,286.50
Provision for Bad Debts	25,015.61
Capital Investment	514,792.79
Surplus and Reserve	167,096.92
	$862,440.45

INCOME AND EXPENSES FOR THE TWELVE MONTHS ENDED SEPTEMBER 30, 1923

Income

Membership Dues		$230,183.64
Interest		5,447.07
Initiation Fees		33,076.77
Advertising (Mechanical Engineering)	$141,164.13	
	77,517.53	
		63,646.60
Condensed Catalogues	79,425.90	
	51,311.63	
		28,114.27
Meetings Registration Fees		1,429.00
A.S.M.E. News		4,455.00
Sales — Miscellaneous	7,649.73	
	6,795.70	
		854.03
Sales — Publications	39,357.80	
	34,335.96	
		5,021.84
Sales — Transactions Binding Vol. 43		4,750.00
Sales — Transactions Binding Vol. 44		4,464.00
Total		$381,442.22

Expenses

Meetings	$ 19,269.58	
Council and Contingencies	14,639.50	
Federated American Engineering Societies	17,200.71	
Publications:		
Mechanical Engineering & A.S.M.E. News	81,688.70	
Transactions — Vol. 43	35,996.74	
Transactions — Vol. 44	14,000.00	
Local Sections	47,629.47	
Society Development	4,132.28	
Membership Applications	9,176.16	
Student Branches	3,017.06	
Professional Divisions	4,012.19	
Standardization	7,492.94	
Power Test Codes	5,998.44	
Boiler Code	7,665.92	
American Engineering Standards Committee	1,500.00	
Nominating Committee	1,475.84	
Constitution and By-Laws, and Code of Ethics	1,683.28	
Group Insurance	645.57	
Research	3,097.25	
Accounting	11,507.64	
Administration	18,738.35	
House	509.90	
Rent (U. E. S.)	9,974.99	
Employment Service Bureau	4,180.36	
Publicity	2,351.53	
Miscellaneous	4,300.53	
Library	8,003.30	
Provision for bad debts	25,015.61	
		$364,903.84
Excess of Income over Expenses		$ 16,538.38

APPENDIX NO 2

CONSTITUTION

ARTICLE C1 — NAME AND GOVERNMENT

SECTION 1. The name of this Society is The American Society of Mechanical Engineers.

SECTION 2. The Society is a corporation, organized April 7, 1880, and chartered under the laws of the State of New York, December 23, 1881. A supplemental charter was issued on October 17, 1907, when the Society was consolidated with the Mechanical Engineers' Library Association.

The principal offices of the Society shall be in the City of New York.

SECTION 3. The Society shall be governed by this Constitution, the By-Laws, and the Rules.

ARTICLE C2 — OBJECTS

SECTION 1. The objects of this Society are to promote the art and science of mechanical engineering and the allied arts and sciences; to encourage original research; to foster engineering education; to advance the standards of engineering; to promote the intercourse of engineers among themselves and with allied technologists; and severally and in coöperation with other engineering and technical societies to broaden the usefulness of the engineering profession.

ARTICLE C3 — MEMBERSHIP

SECTION 1. The membership shall consist of Honorary Members, Members, Associates, Associate-Members and Juniors.

SECTION 2. The rights and privileges of every member shall be personal to himself and shall not be transferable.

SECTION 3. Each member shall be entitled to vote on any question before any meeting of the Society, or before the Society as a whole.

SECTION 4. Every person admitted to membership shall be subject to the Constitution of the Society, and to any amendments that may be made from time to time.

ARTICLE C4 — QUALIFICATIONS FOR ADMISSION

SECTION 1. Members of all grades shall be elected by the Council.

SECTION 2. An Honorary Member shall be a person of acknowledged professional eminence.

SECTION 3. A Member shall be an engineer, at least thirty-two (32) years of age, who has been in the active practice of his profession, or who has fulfilled the duties of a professor of engineering in a college or school of accepted standing, for at least ten (10) years, and has been in responsible charge of important work for at least five (5) years, and is qualified to design as well as to direct engineering work.

Graduation from a school of engineering of accepted standing shall be considered equivalent to two (2) years of active practice.

SECTION 4. An Associate need not be an engineer, but must have had such responsible connection with some branch of engineering, science, the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge, and he must be at least thirty (30) years of age.

SECTION 5. An Associate-Member shall be an engineer, at least twenty-seven (27) years of age, who has been in the active practice of his profession,

or who has fulfilled the duties of a professor of engineering in a college or school of accepted standing, for at least six (6) years, and has been in responsible charge of work for at least two (2) years.

Graduation from a school of engineering of accepted standing shall be considered equivalent to two (2) years of active practice.

SECTION 6. A Junior must have had such engineering experience as will enable him to fill a subordinate position in engineering work, or he must be a graduate of an engineering school of accepted standing. He must be at least twenty-one (21) years of age, and his connection with the Society shall cease when he becomes thirty (30) years of age unless he be previously transferred to another grade.

ARTICLE C5 — FEES AND DUES

SECTION 1. The initiation fee for membership in each grade shall be:

Member	$25
Associate	25
Associate-Member	25
Junior	10
Promotion from Junior to a higher grade	15

SECTION 2. The annual dues for membership in each grade shall be:

Member	$15
Associate	15
Associate-Member	15
Junior:	
for the first six (6) years of his membership as a Junior	10
and thereafter	15

SECTION 3. The Council may permit any Member, Associate or Associate-Member to become a Life-Member in the same grade, as provided in the By-Laws.

SECTION 4. The Council may remit the dues of any member for any special reason, as provided in the By-Laws.

ARTICLE C6 — NOMINATING COMMITTEES

SECTION 1. The membership of the Society shall elect annually a Regular Nominating Committee, whose duty shall be to select candidates for the elective offices to be filled at each annual election, as provided in the By-Laws.

SECTION 2. Other nominating committees having the same powers may be constituted by the membership of the Society, as provided in the By-Laws.

ARTICLE C7 — DIRECTORS

SECTION 1. The affairs of the Society shall be managed by a board of directors, chosen from its membership and styled "The Council."

SECTION 2. The Directors of the Society shall consist of a President, six (6) Vice-Presidents, nine (9) Managers, the last five (5) surviving Past-Presidents, and a Treasurer.

SECTION 3. The Directors shall be elected at the Annual Meeting of the Society, on the first Tuesday in December, as provided in the Charter.

The election shall be by sealed letter-ballot of the membership, as detailed in the By-Laws.

SECTION 4. The President shall be elected for one (1) year, the Vice-Presidents for two (2) years, the Managers for three (3) years, and the Treasurer for one (1) year.

SECTION 5. The Officers of the Society shall consist of the President, the Vice-Presidents, and the Treasurer.

SECTION 6. The Directors may at any time, whenever sufficient cause

shall appear to them, delegate to any member of the Society the performance of any duties required by the Constitution to be performed by any Director or by the Secretary.

ARTICLE C8 — COUNCIL

SECTION 1. The Council shall have full control of the activities of the Society, subject to the limitations of the Constitution.

SECTION 2. The Council shall have power to fill vacancies in its membership by appointment until the next election, as provided in the By-Laws, except that the office of President shall be filled by the Vice-President who is senior by age.

SECTION 3. The number of members constituting a quorum of the Council shall be as determined in the By-Laws.

SECTION 4. The Council shall present at the Annual Meeting of the Society a report verified by the President or Treasurer or by twelve (12) members of the Council, showing the whole amount of real and personal property owned by the Society, where located, and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated or expended during the year immediately preceding such date, and the purpose, object or persons to or for which such applications, appropriations, or expenditures have been made; also the names and places of residence of the persons who have been admitted into membership in the Society during the year.

The report shall be filed with the records of the Society, and an abstract shall be entered in the minutes of the proceedings of the Annual Meeting of the Society.

ARTICLE C9 — MEETINGS OF THE SOCIETY

SECTION 1. The Annual Meeting of the Society shall be held at such time and place as the Council shall appoint, provided it begins in the City of New York and continues there during the annual election of Directors, held on the first Tuesday in December.

SECTION 2. The Semi-Annual Meeting of the Society shall be held at such time and place as the Council shall appoint, as provided in the By-Laws.

SECTION 3. A Special Meeting of the Society may be called at any time and place at the discretion of the Council, or shall be called by the Council upon the written request of at least one (1) per cent of the membership.

The call for the meeting shall be issued at least thirty (30) days prior to the date set for it, and shall state the business to be considered. No other business shall be transacted at the meeting.

SECTION 4. The number of members constituting a quorum at any meeting of the Society shall be as determined in the By-Laws.

SECTION 5. An action of a Meeting of the Society shall be deemed an action of the Society as a whole. Any expenditure required by such action is subject to approval and authorization by the Council.

ARTICLE C10 — PROFESSIONAL DIVISIONS

SECTION 1. The Council may authorize the organization of Professional Divisions composed of members of any or all grades, which shall operate under the provisions of the Constitution, By-Laws and Rules.

ARTICLE C11 — LOCAL SECTIONS

SECTION 1. The Council may authorize the organization of Local Sections composed of members of any or all grades, which shall operate under the provisions of the Constitution, By-Laws and Rules.

ARTICLE C12 — PUBLICATIONS AND PAPERS

SECTION 1. The papers and publications of the Society shall be issued in such manner as the Council may direct.

ARTICLE C13 — SECRETARY

SECTION 1. At its first meeting after the Annual Meeting of the Society the Council shall appoint a member of the Society to serve as Secretary for one (1) year.

SECTION 2. The Secretary shall perform the duties usually pertaining to this office, in accordance with the By-Laws and Rules, and such further duties as may be required by the Council.

SECTION 3. Any vacancy in the office of Secretary shall be filled by appointment by the Council.

ARTICLE C14 — FUNDS

SECTION 1. The deposit, investment, and disbursement of all funds shall be subject to the direction of the Council.

ARTICLE C15 — PROFESSIONAL PRACTICE

SECTION 1. In all professional and business relations the members of the Society shall be governed by the Code of Ethics incorporated in the By-Laws.

SECTION 2. Any member who has violated the Constitution of the Society, or who is guilty of conduct rendering him unfit to remain a member, may be expelled by the vote of fifteen (15) members of the Council, after he has been given opportunity to be heard in his own defense.

SECTION 3. The Society may approve or adopt any report, standard, code, formula, or recommended practice.

SECTION 4. The Society shall forbid and oppose the use of its name or initials in any commercial work or business, except to indicate conformity with its standards or recommended practices, in accordance with the By-Laws and Rules.

ARTICLE C16 — AMENDMENTS

SECTION 1. At any Meeting of the Society, any person entitled to vote may propose in writing an amendment to this Constitution, provided that it shall bear the written endorsement of at least one (1) per cent of the membership.

Such proposed amendment shall not be voted on for adoption at that meeting, but shall be opened to discussion and modification, and to a vote as to whether, in its original or modified form, it shall be mailed in printed form to the members of the Society for action.

If the members present at the meeting, not less than twenty (20) voting in favor thereof, shall so decide, then the Secretary shall mail in printed form to each person entitled to vote, at least sixty (60) days previous to the next Meeting of the Society, a copy of the proposed amendment as so decided by said vote, accompanied by any comment the Council may elect to make.

A ballot shall be sent with the proposed amendment, and the voting shall be by sealed letter-ballot, closing at noon of the twentieth (20th) day preceding the Meeting of the Society following the mailing.

The ballots shall be voted, canvassed and announced as provided in the By-Laws.

The adoption of the amendment shall be decided by a majority of the votes cast.

The Presiding Officer at the Meeting of the Society following the close

of the ballot shall announce the result, and if the amendment is adopted it shall thereupon take effect.

SECTION 2. Any changes in the order or numbering of articles or sections of the Constitution required by an amendment shall be made under the direction of the Council.

SECTION 3. This Constitution shall supersede all previous rules of the Society, and shall go into effect upon the adjournment of the Meeting of the Society at which the Presiding Officer announces its adoption.[1]

CODE OF ETHICS

Engineering work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The Engineering Profession is held responsible for the planning, construction, and operation of such work, and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1 The Engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country, and devotion to high ideals of courtesy and personal honor.

2 He will refrain from associating himself with, or allowing the use of his name by, an enterprise of questionable character.

3 He will advertise only in a dignified manner, being careful to avoid misleading statements.

4 He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5 He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6 He will refrain from using any improper or questionable methods of soliciting professional work, and will decline to pay or to accept commissions for securing such work.

7 He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8 He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9 He will coöperate in upbuilding the Engineering Profession by exchanging general information and experience with his fellow-engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science, and the technical press.

10 He will interest himself in the public welfare, in behalf of which he will be ready to apply his special knowledge, skill, and training for the use and benefit of mankind.

[1] Put into effect at the Annual Meeting of the Society, December, 1922.

No. 1902

THE SALT VELOCITY METHOD OF WATER MEASUREMENT

By Charles M. Allen,[1] Worcester, Mass.
Member of the Society
and
Edwin A. Taylor,[2] Worcester, Mass.
Non-Member

This paper describes a new method of water measurement called the Salt Velocity Method. The authors outline the theory and development of the method, describe the apparatus and methods of computation used in laboratory and field tests, give an account of several commercial tests, and present for discussion the claims of the method for a high degree of accuracy and reliability.

INTRODUCTION

During the past few years the engineering societies and the technical journals have published many articles and discussions on the measurement of water power and on the design, tests, and efficiency of water wheels and settings. But very little has been published recently dealing with the measurement of water in flumes, pipes, and penstocks, and particularly with the reliability and accuracy of various methods and their applicability to the field testing of modern or recently designed water-power units.

2 The most accurate known method of measuring water is by weighing, but this method is limited to comparatively small quantities. Other methods which have been used with varying degrees of accuracy are floats, weir, current meter, pitot tube, venturi meter, color velocity, moving screen, chemical method, and the Gibson method.

3 Measuring water by means of floats in the canal is sometimes done, but conditions are seldom found where accurate results can be obtained by this method.

4 The weir, properly designed and used, will indicate the dis-

[1] Professor of Hydraulic Engineering, Worcester Polytechnic Institute.
[2] Hydraulic Engineer, Worcester, Mass.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers. For discussion see p. 377.

charge within one or two per cent of the true quantity, but on account of loss of head and construction costs and the difficulties in obtaining uniform velocities of approach, the weir is usually not practical. If a weir can be calibrated by another more accurate method, as is sometimes possible in laboratories, it then becomes as accurate as the check method.

5 The current meter can be used in open channels of some power plants, preferably in the forebay and in the tailrace only when quiet water conditions prevail. If water conditions are good and the meter is properly rated and used, it should give results well within commercial requirements.

6 The pitot tube in various forms can usually be used at a reasonable cost in wood or steel penstocks. The pitometer, which consists of a pair of pitot tubes and is capable of being reversed, is an accurate and convenient form of this instrument.

7 The venturi meter is accurate, but unless the meter has already been installed, its cost would prohibit the installation for tests alone.

8 The color velocity method is only applicable to fairly long pipe lines. It appears to be very accurate, but up to date comparatively few tests have been made with this method, and as yet no means of recording graphically the passage of the color are available.

9 The moving screen, although very accurate, is applicable to permanent laboratory installations only and has been rarely used in the United States.

10 As so far developed, the chemical method with salt solution is very accurate when properly used, but it is relatively expensive.

11 The Gibson method is a very accurate method in penstocks of considerable length.

12 For many power plants, and particularly the recently designed low-head plants with short concrete penstocks of varying cross-section, all of the above methods are too expensive or too inaccurate, and none of them are universally recognized as a standard for field testing. All of them possess inherent disadvantages under the various conditions present in commercial tests. For some time a growing need has been felt for a simple and accurate method of measuring water under the above conditions and the salt velocity method has been developed to meet those conditions, as well as the conditions found in long penstocks.

Theory

13 The salt velocity method of water measurement is based on the fact that salt in solution increases the electrical conductivity of water. Salt solution is introduced near the upper end of the conduit, and the passage of the solution across one or more pairs of electrodes, at other points in the conduit, is recorded graphically

by electrical recording instruments. The passage of the salt solution between two points is accurately timed, and the volume of the penstock between the same points is accurately determined. The discharge in cubic feet per second equals the volume in cubic feet divided by the time in seconds.

Historical

14 So far as is known by the authors, no successful application of this method of measuring water has ever been made previous to 1921. Early in that year the development of this method was commenced with laboratory tests at the Worcester Polytechnic Institute. In September, 1921, the first commercial tests were made on two units of a power plant. In 1922 the authors conducted extensive investigations at the Alden Hydraulic Laboratory of the Worcester Polytechnic Institute, and at the power plant of the Laurentide Power Company, at Grand Mere, Quebec. In the fall of that year ten successful commercial tests were made. Both the laboratory investigations and the commercial tests have been continued in 1923.

Sections of Paper

15 This paper is divided into six sections, each section covering a certain period of time and a certain group of tests as follows:

I 1921 Laboratory Investigations
II 1921 Commercial Tests
III 1922 Laboratory Investigations
IV 1922 Field Investigations
V 1922 Commercial Tests
VI 1923 Laboratory Investigations

16 Each group of investigations had a common objective, i.e. the determination of the accuracy and reliability of this method of water measurement. But in each group different plants, apparatus, or methods of testing were used, and each group had its own objective independent of the common object. The six groups or sections marked six distinct steps in the development of the method to its present state.

I — 1921 LABORATORY INVESTIGATIONS

Object

17 The object of these tests was to investigate the practical possibilities of the salt velocity method. By visualizing an infinite number of floats equally distributed over the cross-section of the conduit, with each little float recording its own velocity and the whole group automatically recording a composite picture of velocities, the theoretical possibilities of this method can be readily understood.

Plant

18 The plant used for these first tests was Plant No. 1 of the Alden Hydraulic Laboratory of the Worcester Polytechnic Institute, Worcester, Mass. Fig. 1 shows the layout of this plant. The water supply is from two ponds with a combined area of 200 acres, and a constant head can be maintained for long periods. A total fall of 35 ft. is available.

19 The penstock is a riveted steel pipe about 450 ft. long. From the pond to within ten feet of the laboratory the diameter of the pipe is 40 in. In the laboratory are a 36-in. by 16-in. venturi

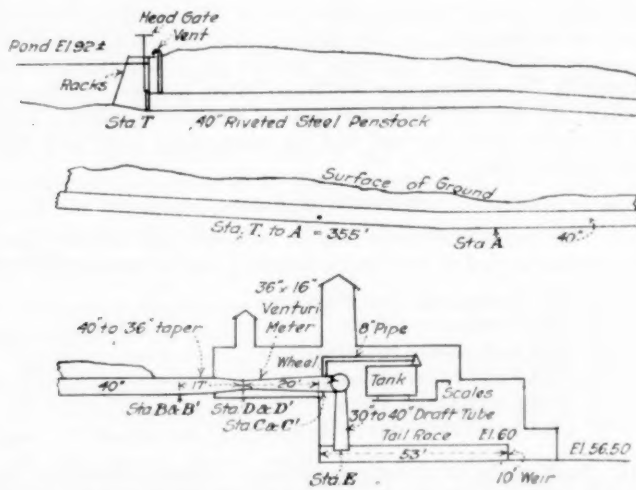


Fig. 1 Sketch of Alden Hydraulic Laboratory and Penstock, Worcester Polytechnic Institute

meter, an 18-in. horizontal water wheel with cylindrical case, and a conical vertical draft tube discharging into a tailrace at the lower end of which is a 10-ft. standard sharp-crested weir with end contractions.

Apparatus and Methods

20 In these tests, as in all subsequent tests, the apparatus in general consisted of a salt injector and electrodes in the penstock together with signaling and timing devices.

21 *Salt Introduction.* In the very first tests raw salt was introduced into the penstock, the operator punched the stop watch and then ran down the road to the laboratory where an ammeter

was connected to the electrodes, and the operator again punched the watch on deflection of the meter needle.

22 The first improvement in the introduction of salt was a closed metal box containing a charge of salt which was lowered to the mouth of the penstock by a pole. The charge of salt was released when the hinged sides of the box were raised by wires operated from a platform over the head gate. Raw salt or solution was also placed in paper bags tied to a pole, and when lowered to

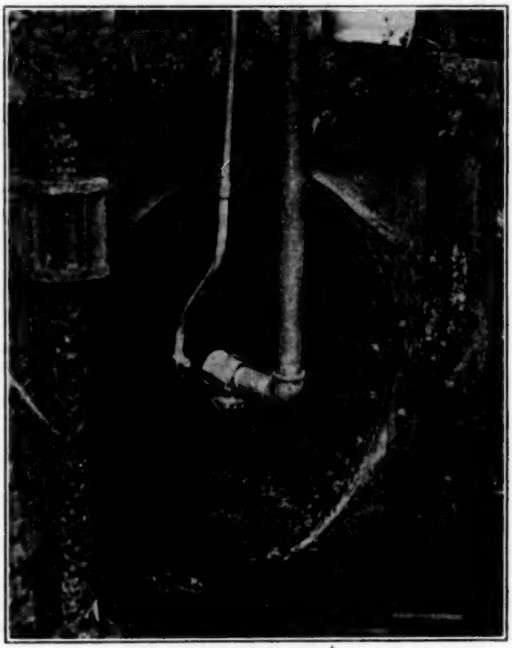


Fig. 2 Head Gate, Mouth of Penstock and 2-in. Quick-Acting Valve

proper position the bags were broken by a sudden motion of the pole.

23 Later, salt was injected in a solution piped from an elevated mixing tank over the head gate. A 2-in. pipe led to a quick-acting valve, operated by a rod from the surface platform. The valve was fitted with a vertical deflecting plate and scavenging tubes and was placed facing downstream in the plane of the penstock entrance. The strength of the charge could be varied by changing the degree of saturation of the solution or by throttling at a slow-motion valve in the feed pipe. Fig. 2 shows the head gate, pipe, and valve.

24 *Electrodes.* The first pair of electrodes used were thin strips of copper, 38 in. long and 2 in. wide, spaced 2 in. apart by wooden blocks. These electrodes were placed in a horizontal position across the center of the pipe and held in place by wooden wedges. A similarly constructed electrode 6 in. long was used for traversing the pipe. This electrode was fastened to a rod passing out of the pipe through a stuffing box and could be held at any position along the diameter of the pipe. Usually during a traverse this small electrode would be held in ten different positions.

25 Later, several electrodes were made of thin copper strips $\frac{3}{4}$ in. wide and 4 in. long, spaced $\frac{3}{16}$ in. apart. These electrodes were attached to short pitometer rods, rubber-covered wires being substituted for the original pitot tubes. The rods were packed and then screwed to nipples in front of gate valves. This form of electrode could be placed in any position across the pipe, rotated to free itself from debris, or withdrawn entirely. Fig. 3 shows a 6-in. and a traversing electrode.

Fig. 3 Electrodes Used in Laboratory, 1921

26 *Meters.* All electrodes were connected by wires and switches to the indicating meters. In 1921 only indicating voltmeters and ammeters were used to record the current between the electrodes. Direct current at 110 volts was used on the circuits.

27 *Timing.* Soon after the first few tests had been made, a telephone line between the pond and laboratory replaced the running operator. The stop watches were then started by the operator at the laboratory on verbal signal from the pond operator, and the time was observed at various stages of the needle deflection. As a rule, two watches were used. The first watch was stopped on the initial appearance of the salt, the second watch at final appearance, and the intermediate stages were observed by metronome with the second watch running.

28 *Standard of Measurement.* The standard of measurement used in these tests was the 10-ft. weir at the lower end of the laboratory. Two hook gages in isolated stilling boxes at either side of the tailrace indicated the heads on the weir. The zero point on these gages and the level of the weir were frequently tested.

29 *Number of Tests.* Including trials, about 400 charges of salt solution (one charge called a "shot") were used in these tests, which were grouped into 30 runs of from 10 to 20 shots at each gate opening. Runs at each gate opening were repeated six to eight times.

Computations

30 During the 1921 series of laboratory tests, the computation of the discharge by the salt velocity method for comparison with the quantity indicated by the weir was made by three different methods:

1 The time was computed from the moment of salt introduction to the initial appearance of the salt at the electrodes, i.e., the beginning of the curve, and a coefficient 1.095 was computed to give the true Q. This coefficient remained constant for all gate openings and velocities.

2 The time was computed from the moment of salt introduction to the point of mean time between the initial and final appearance of the salt at the electrodes, i.e., half-way between the beginning and end of the curve, and coefficients were computed to give the true Q. These coefficients varied and no constant could be established.

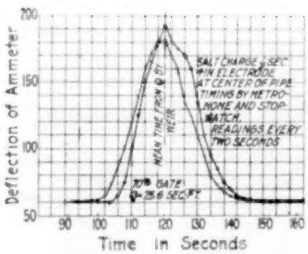


Fig. 4

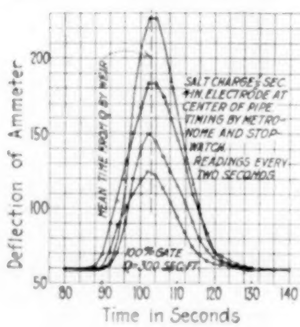


Fig. 5

Sample Curves from Tests on 40-in. Pipe, 355 Ft. Long

3 The time was computed from the moment of salt introduction to the moment of maximum density of the salt solution passing the electrodes, i.e., the point of maximum deflection of the needle or the peak of the curve, and a coefficient was computed to yield the true Q by weir. This coefficient was approximately 1.00 (averaged 0.9975).

31 The length and diameters of the section of pipe used were carefully measured and the volume of the section computed. Then

$$\frac{\text{Vol. (cu. ft.)}}{\text{Time (sec.)}} = Q \text{ (cu. ft. per sec.)}$$

32 The quantity by weir was computed by the Francis formula, namely, $Q = 3.33\ (b - 0.2h)h^{3/2}$. The heads over the weir crest varied from six inches to one foot.

RESULTS

33 The results of computations by each of the foregoing methods are given in Tables 1 and 2. The series of tests in Table 1 were all made by injecting solid salt through the metal box. This table was taken from a thesis by Messrs. Bijur and Scanlan of the class of 1921, Worcester Polytechnic Institute.

TABLE 1 RESULTS COMPUTED FROM TIME OF INITIAL APPEARANCE OF SALT AND FROM MEAN OF TOTAL APPEARANCE

Q by weir, second-feet	Q computed, 1.095 × initial time	Variations, per cent	Q computed, 0.865 × mean time	Variations, per cent	Q computed, 0.900 × mean time	Variations, per cent
6.4	6.4	0.0	7.3	+ 14.1	7.0	+ 9.4
13.6	13.4	− 1.5	13.7	+ 0.7	13.2	− 2.9
18.1	18.3	+ 1.1	18.1	0.0	17.4	− 3.9
21.4	21.4	0.0	21.4	0.0	20.6	− 3.7
26.3	26.5	+ 0.8	26.3	0.0	25.3	− 3.8
32.0	32.0	0.0	30.6	− 4.4	29.6	− 7.5
35.6	35.7	+ 0.3	32.6	− 8.4	31.4	− 11.8
21.8	21.8	0.0	23.4	+ 7.3	22.5	+ 3.2
24.9	24.9	0.0	26.6	+ 6.8	25.6	+ 2.8
29.9	29.9	0.0	31.8	+ 6.4	30.6	+ 2.3
31.6	31.3	− 1.0	Traverse			
32.0	32.0	0.0	33.3	+ 4.1	32.0	0.0

TABLE 2 RESULTS COMPUTED FROM TIME OF MAXIMUM DENSITY OF SALT

Solid Salt or Solution in Paper Bags			Salt Solution Through Pipe and Valve		
Gate opening, per cent	Q by weir	Coefficient = $\frac{\text{Max. time by salt}}{\text{True time by weir}}$	Gate opening, per cent	Q by weir	Coefficient = $\frac{\text{Max. time by salt}}{\text{True time by weir}}$
20	13.4	1.008	40	18.3	0.998
40	18.8	1.002	40	18.2	0.989
60	22.8	0.994	20	14.4	0.989
100	29.2	1.000	40	19.6	0.999
		Avg. 1.001	60	23.9	0.995
			80	27.3	0.999
			100	28.9	1.005
			80	27.6	0.998
			30	17.6	0.995
			50	22.3	0.998
			80	27.4	1.002
			30	16.9	0.990
			50	21.8	0.990
			70	25.6	1.000
			100	30.0	1.001
					Avg. 0.997

34 Figs. 4 and 5 show curves with the meter deflection plotted on time in seconds. Fig. 6 shows a curve of true time, as determined by the weir, plotted on discharge, with the results by the salt velocity method spotted in circles and crosses. Fig. 7 shows curves illustrating the percentage of the true Q (by weir) obtained by computing the salt velocity results, using the time of initial appearance of the salt, the mean between the initial and final appearance, and the time of maximum density of the salt. During the tests the interior of the pipe was cleaned, and the results are shown both before and after cleaning.

Conclusions

35 These first laboratory tests showed conclusively that the tests could be made and repeated indefinitely with consistent results. It showed an apparently constant relation between the initial appearance of the salt at the electrodes and the time of maximum density of the salt passing the electrodes. When properly computed the discharge by the salt velocity method checked the true Q by weir within about one per cent for single runs, and much closer for a long series of runs.

36 Prior to and during these laboratory tests on salt velocity a number of tests were made by the color velocity method. A

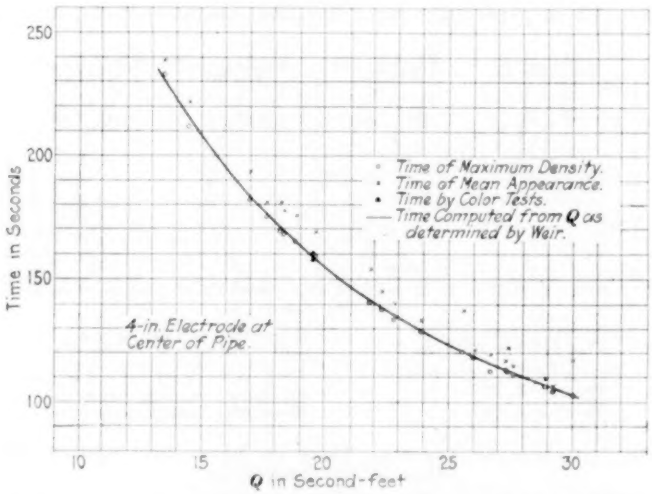


FIG. 6 CURVE OF TESTS ON 40-IN. PIPE. TRUE TIME AS DETERMINED BY WEIR

strong solution of a red aniline dye was enclosed in paper bags which were broken at the penstock entrance, and stop watches were used for timing. At the lower end of the penstock an open-ended 2-in. glass pipe was used to observe the appearance of the color. This glass was arranged with a white background. By taking the *mean* time between the initial and final appearance of the color the results checked very closely with the results by salt velocity. (See Fig. 6.)

37 The reason why the *mean* time gave accurate results by color but not by salt was because with color the eye cannot detect the color when well diluted with penstock water. If a curve of the real color appearance could be made, the portion of the curve visible to the eye would be approximately the upper third only,

and this portion of the curve is symmetrical. With salt the whole curve is visible and the two ends are usually not symmetrical.

II — 1921 COMMERCIAL TESTS

38 In September, 1921, tests were made of two hydroelectric units in New Hampshire. The object of these tests was to determine the discharge of each unit at various gate openings. Each unit tested had a penstock of 13-ft. wood-stave pipe 1400 ft. long.

Apparatus and Methods

39 *Salt Introduction.* A few tests were made by introducing raw salt in paper bags as in the first laboratory tests, but the

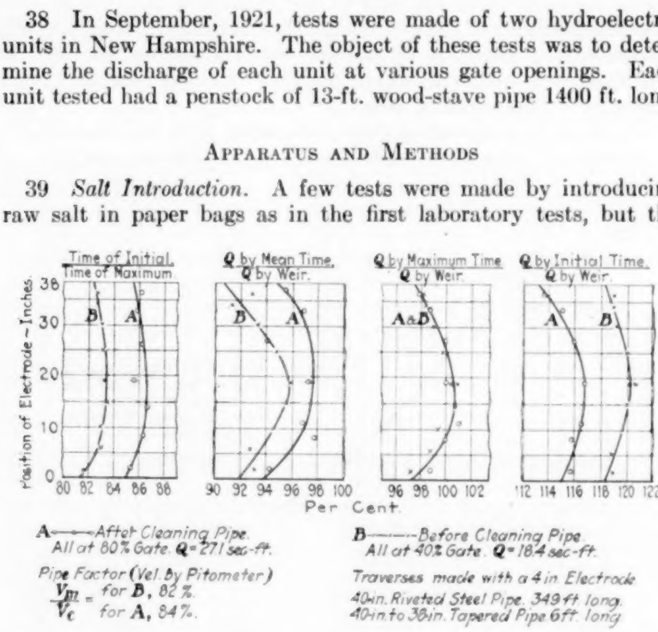


Fig. 7 Curves of Traverse Tests on 40-in. Pipe, with Various Methods of Computation

majority of the tests were made by wrapping the raw salt in a cotton sheet tied to a rod. When the sheet was lowered to the proper position in the mouth of the penstocks, a rope was pulled, allowing the sheet to open, and the salt was released.

40 *Electrodes.* The electrodes, made of thin copper plates 1 in. by 16 in. spaced $\frac{1}{4}$ in. apart, were fastened to an insulated wooden plug driven into a $1\frac{1}{4}$-in. pipe which was inserted in the penstock through a stuffing box. This electrode was located at one-third the diameter in the pipe. An indicating ammeter was used to indicate the passage of the salt across the electrode.

41 *Timing.* The operators at the salt station and at the meter were connected by telephone, and as the first operator released the salt, he signaled the second operator, who started three stop watches simultaneously.

42 By carefully watching the meter needle, one watch was stopped at the initial appearance of the salt, the second watch at

the maximum deflection of the needle, and the third watch was stopped at the completion of the passage, when the needle returned to normal position. The watches were calibrated before, during, and after the tests.

43 *Standard of Measurement.* No accurate standard of measurement was used for comparison during the tests, but after the tests two current meters were used and two tests made at the same gate opening, 70 per cent.

44 *Computations.* All times were computed from the moment

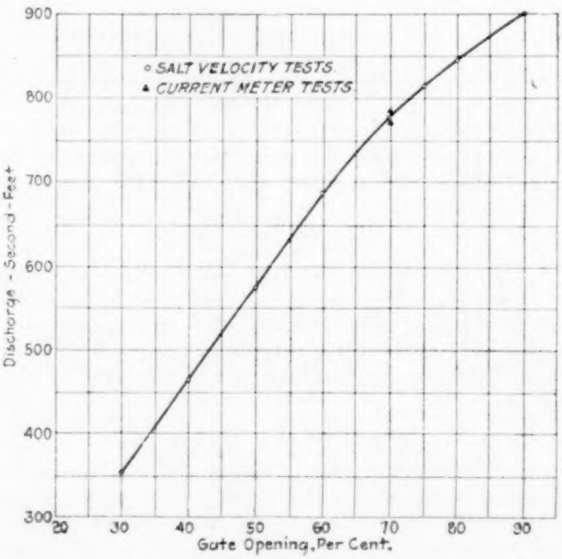


FIG. 8 CURVE OF DISCHARGE OF 13-FT. WOOD-STAVE PIPE, 1400 FT. LONG, UNDER 66 FT. HEAD

of salt introduction, i.e., from the release of the salt charge to the moment of maximum density of the salt solution passing the electrode, i.e., the maximum deflection of the meter needle.

RESULTS

45 The results of these tests were very satisfactory. The work was easily done with very simple apparatus. By reducing to a common head and plotting the discharge against gate opening, smooth curves were the result.

46 Fig. 8 shows a curve of discharge on gate opening. This curve passes through every test point. The two check tests by

current meter are shown at 70 per cent gate, and the salt velocity curve passes midway between them. These two current-meter tests varied from each other by 2 per cent.

Conclusions

47 These tests confirmed the results of the laboratory tests and showed that the salt velocity method of water measurement was applicable to power plants with long penstocks of uniform diameter.

III — 1922 LABORATORY INVESTIGATIONS

48 After learning of the development of the salt velocity method during 1921, Mr. John Riddile, Chief Engineer for the Laurentide Power Company of Grand Mere, Quebec, was convinced of the possibilities of the method for testing their units. During 1922 an extensive series of investigations was conducted for that company.

Object

49 Up to then the method had been used only on pipes of uniform diameter. This company's power house has short, rectangular, converging penstocks, and the object of this series of tests was to determine the accuracy and applicability of the salt velocity method of water measurement to that type of penstock.

Apparatus and Methods

50 *Plant.* These tests were conducted at the Worcester Polytechnic Institute Laboratory, but instead of using the 40-in. pipe all the time as in 1921, a majority of the tests were made on the pipe line below the 40-in. section, i.e., through the converging portion of the penstock, through the venturi meter, and through the wheel and draft tube. Fig. 9 shows the 36-in. by 16-in. venturi meter.

51 *Salt Introduction.* A few tests were made introducing salt at the pond with the same elevated salt-mixing tank, pipe, and quick-acting valve as were used in the 1921 tests. The remaining tests were made by introducing salt just below the lower end of the 40-in. section of penstock. This point was station B. (See Fig. 1.) The apparatus installed here consisted of an elevated 100-gal. mixing tank piped to a 15-gal. pressure tank, which in turn was piped to the penstock by $\frac{3}{4}$-in. hose and pipe. An air pipe from a pump and storage tank in the laboratory was connected to the pressure tank. Air pressure up to 60 lb. was available. Besides the air pipe and the feed pipe, the pressure tank was also fitted with screens, an inlet pipe, a waste pipe, an air vent, and a pressure gage.

52 Slow-motion valves were placed on all connections, and on

the discharge pipe a $\frac{3}{4}$-in. quick-acting valve was placed near the penstock. The brass distribution pipe was inserted into the penstock through a stuffing box. Various ways of discharging the brine from the pipe were tried: a $\frac{3}{4}$-in. open end; a $\frac{1}{4}$-in. open end; two $\frac{1}{4}$-in. holes at the center of the pipe; and 18 holes of graded sizes extending on both sides of the pipe. These holes were arranged so as to discharge vertically. This form of distribution was called a "perforated introduction." During the latter portion of these tests a $\frac{3}{4}$-in. pop valve was attached to the end of the pipe for the introduction of salt at station B. This pop valve opened and closed under pressure from the feed pipe, which was controlled by the quick-acting valve.

53 *Electrodes.* Various electrodes were used. A double-helix electrode 4 in. long was made of copper wire wound around a $\frac{3}{4}$-in.

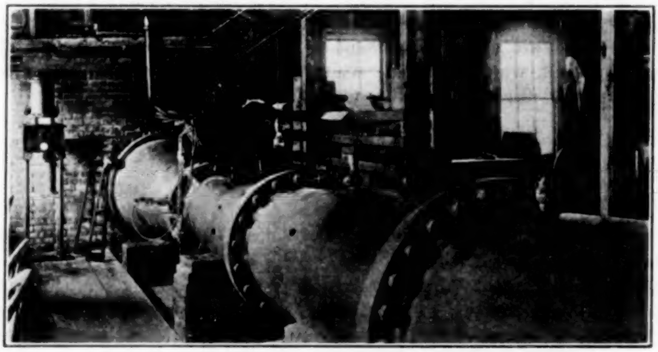


FIG. 9 36 x 16-IN. VENTURI METER AT ALDEN LABORATORY

wooden plug fitted into the end of a $\frac{3}{4}$-in. pipe. Plate electrodes were made of strips of copper $\frac{1}{16}$ in. thick and 1 in. wide, and from 1 in. to 8 in. long, with spacings of from $\frac{1}{8}$ in. to $\frac{1}{4}$ in. between strips. All of these electrodes were fastened to pitometer rods and could be adjusted through stuffing boxes at any distance along the diameter of the pipe. This form of electrode is shown in Fig. 3. Other electrodes were made of 1-in. brass pipe, placed parallel, extending across the penstock and fastened to the walls by wooden insulating wedges. These pipes were spaced $\frac{3}{4}$ in. apart.

54 In connection with these pipe electrodes was an electrode called a "saddle." This was a strip of $\frac{1}{16}$-in. copper, 1 in. wide and 6 in. long, fastened to and insulated from the center of the upper brass pipe with a $\frac{1}{4}$-in. space between.

55 *Meters.* The electrodes were wired to a portable Bristol direct-current recording ammeter. This instrument had two meters with a capacity of three amperes each. Direct current at

110 volts was used. The roll of paper, or chart, for this meter was motor driven.

56 The passage of the brine by the different electrodes was recorded on the chart by two pens, each actuated by its own meter. The salt introduction was recorded by means of a snap switch at station T (Fig. 1), which was operated by hand simultaneously with the opening and closing of the introduction valve. At station B an automatic contact switch was placed on the handle of the introduction valve. These switches were wired to the ammeters with lamps in series for resistance, recording the time and duration of the introduction of the salt by either one of the pens, just mentioned, on the same chart.

57 *Timing.* A standard seconds-pendulum clock was wired to the ammeter, and by means of a magnet and relay recorded seconds by a separate pen (a third pen) on the same chart. For convenience in counting seconds, a break occurred every minute, at which time the pen missed two seconds records.

58 *Standard of Measurement.* The standard for the water measurement was the 10-ft. weir. The venturi meter was frequently used for check measurements.

Description of Tests

59 The general arrangement for these tests was an observer and operator at the laboratory who arranged the electrodes, operated the motor, ammeter, and switches, and took readings on the weir and venturi meter. This observer was connected by telephone to an operator at station T when salt was introduced at the pond. This second operator prepared the brine and operated the salt valve and the introduction-signal switch. When salt was introduced at station B, near the laboratory, the operators were near enough to talk. If a third operator was present, he prepared the brine, connected the air pressure, and read the gages for weir and venturi meter.

60 All tests were numbered, and the necessary notes and data were recorded on the chart. In later tests these data were recorded on data sheets, Table 3 being a sample sheet prepared for the early tests.

61 Including trials, about 1200 individual tests or charges of salt solution were used, which were grouped into 60 runs, and these in turn were segregated into 13 groups based on the stations used for the salt introduction and for electrodes. Sample charts illustrating the curves obtained from the various introduction and electrode stations are presented in Figs. 10 and 11. These curves show methods of computation, give comparison of results by salt and by weir, and will be discussed in detail later.

Computations

62 The volumes of the penstock between various stations were computed from surveys.

63 As in 1921, the study of the problem of what point on the curve to use in computing was continued. While recognizing the theoretically correct center of gravity of the curve, the exact determination of that point was still too difficult and required too much time for practical use.

64 The curves made on long sections of pipe continued to be

TABLE 3 SAMPLE DATA SHEET OF SALT VELOCITY TESTS

(Laboratory Investigations, 1922)

Length of steel pipe used {A = 275.85 ft. B = 355.20 ft.} Size, 40 in. Volume {A = 2407 B = 3096}

Run No.	Test No.	Sec. shot	Time of day	Weir	Q Venturi meter	Seconds, T to A	T to A, test average	Seconds, T to B	T to B, test average	Q, cu. ft. per sec. Weir is 100%	Remarks for all runs
	19	1	7/4/22	21.94	....	111.9		140.1		A = 21.58	Salt from tank
	20	1				111.1	111.58	140.9	141.2	−1.6%	at pond
4	21	1				112.7	Q =	141.7	Q =		
	22	2				110.8	21.58	141.7	21.92	B = 21.92	6-in. electrode
	23	2				111.4		141.4		−0.1%	at A and
											12-in. in pipe
	24	1	7/4/22	22.05	....	109.1		140.0		A = 21.96	6-in. electrode
	25	2				109.8	109.56	142.0	140.22	−0.4%	at B and
5	26	2				111.7	Q =	140.4	Q =		12-in. in pipe
	27	3				109.3	21.96	138.4	22.08	B = 22.08	
	28	5				106.9		140.3		+0.1%	110-volt direct current
	29	1	7/13/22 4:30 P.M.	18.10	....	136.0		170.0		A = 17.84	
	30	2				133.0	134.20	170.9	170.92	−1.4%	
6	31	2				133.2	Q =	170.7	Q =		Salt introduced
	32	2				134.3	17.84	171.8	18.12	B = 18.12	through 2-in.
	33	3				134.3		171.2		+0.1%	pipe and 2-in. quick-acting valve
	34	1	7/15/22 5:00 P.M.	20.34	20.50	113.9		153.9		A = 20.72	
	35	1				115.7	116.08	155.8	153.38	+1.9%	
	36	2				115.8	Q =	153.4	Q =		
7	37	2				115.6	20.72	153.3	20.19	B = 20.19	
	38	2				117.4		152.0		−0.7%	
	39	2				118.1		151.9			
	40	2	7/15/22 5:40 P.M.	20.08	20.19	117.7	117.00	153.4	154.00	A = 20.57	
8	41	2				115.9	Q =	152.7	Q =	+2.4%	
	42	3				117.7	20.57	155.4	20.10	B = 20.10	
	43	3				116.7		154.5		+0.1%	

Average percentage for sheet, 5 runs, at A = +0.2 per cent and at B = −0.2 per cent.

symmetrical and the peaks were used with very uniform and accurate results. The curves made when salt was introduced at station B and when a short section of pipe with a varying cross-section was used were not symmetrical. For these curves a few centers of gravity were accurately determined, and for all the remainder the centers of gravity were determined by eye, with fairly uniform and accurate results. For the introduction curves, the point for timing was always taken half-way between the opening and closing of the salt introduction valve.

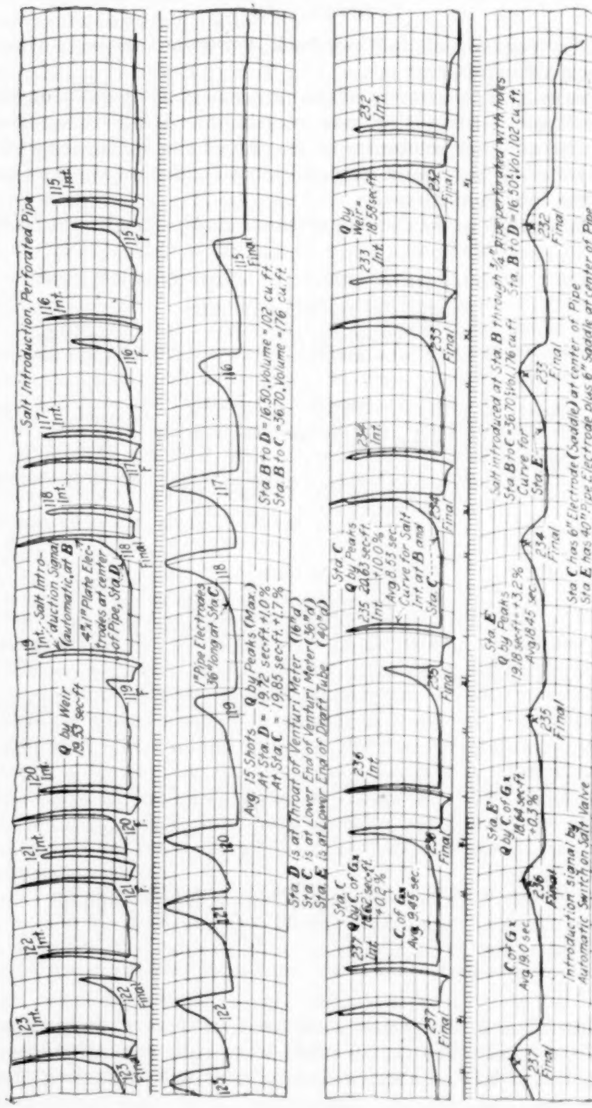


FIG. 10 SAMPLE CURVES, TESTS ON 40-IN. PIPE WITH RESULTS OF DIFFERENT METHODS OF COMPUTATION

65 Sample curves showing the amount of the variation between the maximum deflection or peaks of the curves and the center of gravity of symmetrical and distorted curves are shown in Fig. 12.

Results

66 A summary of the results of these tests is shown in Table 4.

TABLE 4 SUMMARY OF RESULTS OF SALT VELOCITY TESTS

(Investigations at Worcester, 1922. Computed to peaks of curves)

Group No.	From Station	To Station	Length	Salt at	Electrodes	Position	Runs	Shots	Per Cent + (weir = 100%)	Per Cent − (weir = 100%)
1	T	A	275.85	T	4″ helix (wire)	center	1	2	3.50	....
2	T	A	275.85	T	6″ × 1″ plates	12″ in	7	39	....	0.14
3	T	B	355.20	T	6″ × 1″ plates	12″ in	7	39	0.39	....
4	T	B′	355.95	T	1″ pipes, 38″ L.	across	3	20	1.55	....
5	T	C	391.90	T	1″ pipes, 36″ L.	across	3	20	0.13	....
6	B	C	36.70	B	1″ pipes, 36″ L.	across	9	117	0.33	....
7	B	C	36.70	B	6″ × 1″ saddle	center	2	20	4.98	....
8	B	D	16.50	B	4″ × 1″ plates	center	4	48	0.51	....
9	B	D′	17.15	B	4″ × 1″ plates	center	7	102	....	0.13
10	B	E	59.15	B	1″ pipes, 40″ L.	across	2	19	0.92	....
11	B	E	59.15	B	6″ × 1″ saddle	center	5	43	3.38	....
12	B	E	59.15	B	Pipes *and* saddle		4	30	4.79	....
13	T and B	to A, C′, D and D′	16.50 391.15	T B	1″ × 1″ plates, traverse positions	10 to 12	5	325	0.42	....
						Totals	59	824		

Omitting groups 1, 7, 11, and 12 (12 runs and 95 shots) on account of electrodes at center of pipe, and averaging the remainder, shows 47 runs and 729 shots; average + 0.32 per cent.

Averages weighted for number of shots. Q by weir = 100 per cent.

A study of this summary shows that averaging 47 runs with a total of 729 shots gave a quantity for water measurements which was 0.32 of 1 per cent in excess of the quantity measured by the

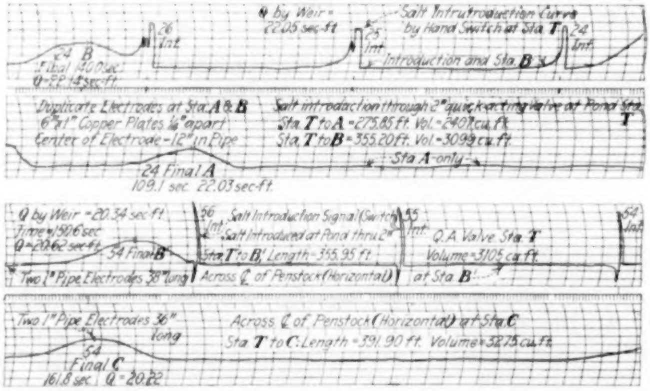


FIG. 11 SAMPLE CURVES, TESTS ON 40-IN. PIPE WITH RESULTS OF DIFFERENT METHODS OF COMPUTATION

weir. Included in these 47 runs are the results with electrodes clear across the pipe, with short electrodes one-quarter to one-third of the diameter into the pipe, with short electrodes in the center of

the pipe at the venturi throat and all traverses across the pipe. The runs omitted from these 47 with accurate results are the runs with short electrodes placed at the center of the pipe at all stations except the venturi throat. The results of these latter runs varied

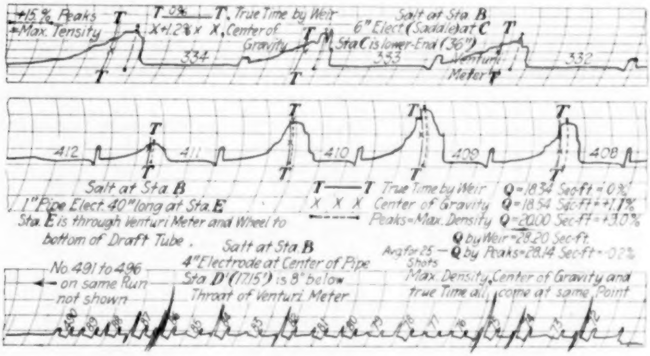


FIG. 12 SAMPLE CURVES, TESTS ON 40-IN. PIPE, SHOWING VARIATION BETWEEN PEAKS AND CENTERS OF GRAVITY OF CURVES

from the true Q by as much as 5 per cent, which is to be expected, since only the fast water at the center of the pipe was measured and all slow water was neglected. (See curves of mean velocities across the pipe which were obtained from the traverses, Figs. 13 and 14.)

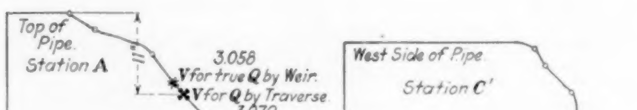


FIG. 13 TRAVERSE CURVES OF AVERAGE VELOCITY FROM SALT VALVE TO ELECTRODES, TESTS ON 40-IN. AND 36-IN. PIPE

67 These traverse curves are not curves of instantaneous velocity at those stations, but are the curves of average velocity from introduction to final electrode. These curves show the effect on Q of using an electrode at the center of the pipe at stations A and C′, and also show that using an electrode at the center at stations D and D′, the throat of the venturi meter, has no appreciable effect. The pipe factor at station D, i.e., the mean

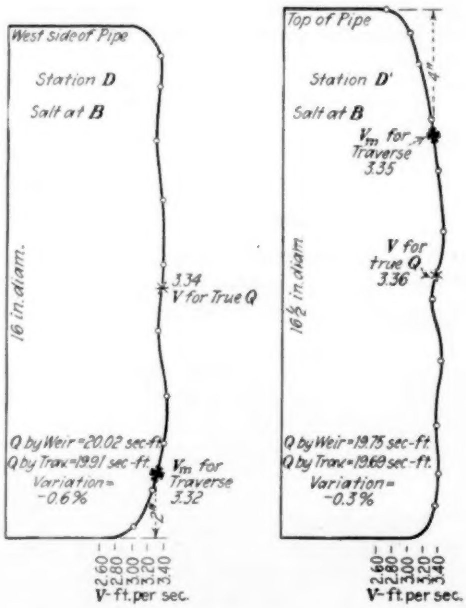


Fig. 14 Traverse Curves of Average Velocity from Salt Valve to Electrodes, Tests on 40-in. and 36-in. Pipe

velocity divided by the center velocity, is 0.994, while at station A the pipe factor is about 0.86 (using instantaneous velocities).

68 These traverses were made with an electrode 1 in. long held at the various points shown by the small circles, and when Q is computed by the equal-area method it varies from the true Q measured on the weir by less than one per cent in each case.

69 Four students of the class of 1923, Worcester Polytechnic Institute, took the salt velocity method of water measurement for their theses. Messrs. Masten and White used the salt velocity method to calibrate a 12-in. by 6-in. venturi meter and an 8-ft. suppressed weir. Messrs. Dodkin and Metcalf made 21 runs with a total of 205 shots on the 40-in. penstock. They introduced salt

at the pond, station T, and used various forms of electrodes at stations A and B. (See Fig. 1.) With the 10-ft. weir as a standard of measurement, the maximum variation of the discharges computed by the salt velocity method were + 1.10 per cent and − 0.95 per cent, with the total 21 runs averaging + 0.10 of one per cent.

Conclusions

70 These tests showed that the results are accurate and reliable for long sections of penstock, with a short electrode inserted

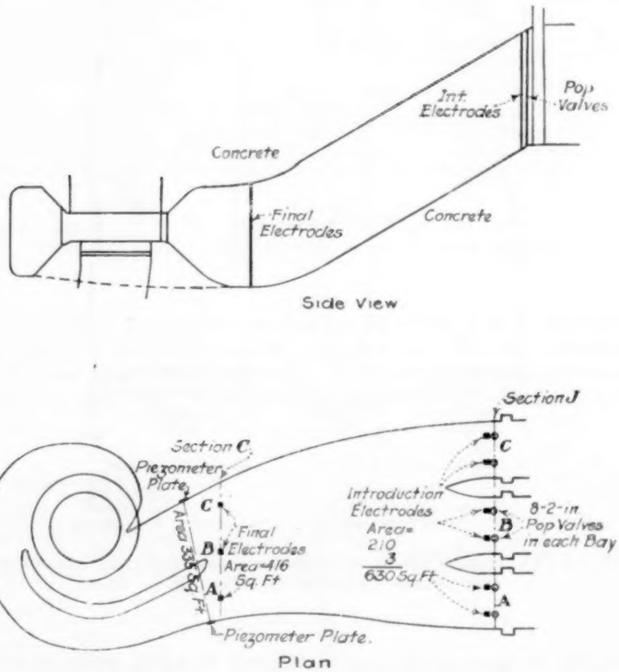


Fig. 15 Sketch of Laurentide Power Company's Penstock

approximately 25 per cent to 30 per cent of the diameter of the pipe. This exact point can be accurately determined by the traverse. The results are also accurate and reliable for all traverses, and for converging or diverging sections of pipe with proper electrodes at proper points in the pipe.

71 Referring to the summary sheet, Table 4, this matter of results with different electrodes at the same station is shown by comparing group 6 with group 7 and group 10 with group 11.

Group 6 at station C and group 10 at station E are with pipe electrodes clear across the pipe and give variations from true Q of $+ 0.33$ of one per cent and $+ 0.92$ of one per cent, respectively, while group 7 and group 11 with 6-in. electrodes in the center of the pipe at the same stations give variations of $+ 4.98$ per cent and $+ 3.38$ per cent from the true Q by weir. Even the parallel pipe electrodes placed across the penstock favored the fast water in the center, thus accounting for the fact that nearly all variations shown on the summary sheet are plus percentages. An improved electrode to correct this feature was used in the 1923 tests at Worcester.

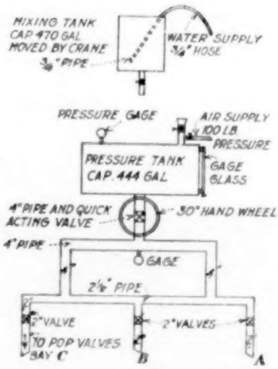


Fig. 16 Sketch of Salt Distribution System

72 The section of the Worcester penstock most nearly approaching the conditions of the Laurentide penstock was from station B to station D or D′, which section includes the upstream

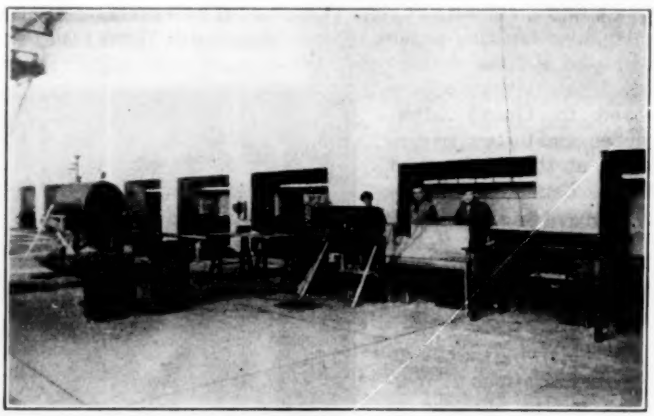


Fig. 17 Apparatus in Gate House, Laurentide Power Company

end of the venturi meter. This section is 39 in. in diameter at B and 16 in. and 16½ in. in diameter at D and D′, respectively. The traverse curves for D and D′ are nearly straight lines. The center water is only slightly faster than the water on the outside, and an electrode at any point in the pipe at these stations gave very accurate results.

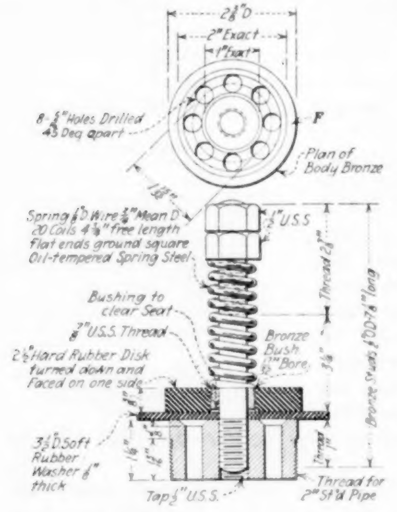


FIG. 18 DETAILS OF POP VALVE

73 The last curve of the sample curves (see Fig. 12) shows some very sharp peaks with no doubt as to their interpretation. These curves were made between stations B and D′ (venturi throat), a distance of 17 ft. The maximum deflection of the peaks and the center of gravity of the curves coincide, and they check with the true quantity as measured by the weir. The average variation for 25 shots was − 0.2 per cent.

74 These 1922 tests at Worcester were so successful and the results were so satisfactory that many doubts concerning the application of the salt velocity method of water measurement to rectangular tapering penstocks were dispelled. These tests were completed and the apparatus in the laboratory was moved to Grand Mere, Quebec, and tests were continued at the power plant of the Laurentide Power Company.

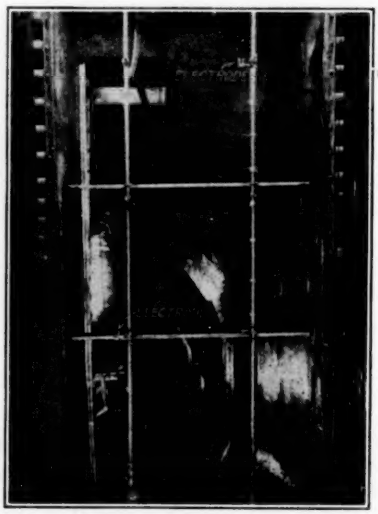


FIG. 19 POP VALVES AND ELECTRODES IN PENSTOCK

IV — 1922 FIELD INVESTIGATIONS

75 Following the laboratory investigations at Worcester in the summer of 1922, field investigations of the salt velocity method of water measurement were made at the power house of the Laurentide Power Co., Ltd., at Grand Mere, Quebec, in October and November, 1922. The object of these tests was to

determine the reliability of the salt velocity method when actually applied to short rectangular tapering penstocks in the field.

Apparatus and Methods

76 *Plant.* Unit No. 7 was used for these investigations. This consists of a 102-in. right-hand water wheel designed by the I. P. Morris Company and built by the Dominion Engineering Works, and connected by a vertical shaft to a generator built by the Canadian Westinghouse Company. The makers' rating of the wheel is 22,000 hp. under 84 ft. head and 120 r.p.m. The penstock is a sloping concrete tube about 65 ft. long, converging from a vertical rectangular cross-section of 630 sq. ft. area at the upper end, to a vertical rectangular cross-section of 335 sq. ft. area at the piezometer plates, which are located at the entrance to the scroll case of the wheel. The upper end of the penstock is divided by concrete piers into three bays of equal area. (See Fig. 15.) In making these tests, from 55 to 57 ft. of penstock (horizontal distance 48 ft.) was used with volumes varying from 27,677 to 28,819 cu. ft.

Fig. 20 Pop Valves in Action under 60 lb. Pressure

77 *Apparatus.* The apparatus used for the salt introduction was installed in the gate house and consisted of a 500-gal. open mixing tank and a 500-gal. pressure tank. The pressure tank was filled by gravity from the mixing tank, which was raised by an overhead traveling crane. A 4-in. pipe led from this pressure tank through a header and equalizer to three 2½-in. leader pipes and three lengths of 2½-in. fire hose, one hose passing into each bay at the upper end of the penstock. Fig. 16 is a diagram and Fig. 17 a photograph showing the arrangement of these tanks and pipes.

78 Each length of hose was connected to eight pop valves of special design located in each bay of the penstock. These 24 pop valves were arranged so as to give a uniform distribution of salt over the entire cross-section. The distribution of brine to the pop valves was controlled by a 4-in. quick-acting valve which had a 30-in. handwheel near the pressure tank. The brine was

also controlled by valves at the upper end of each leader pipe. Figs. 18 and 19 show the details of construction and location of these pop valves. Fig. 20 shows the pop valves in bay C in action while the penstock was emptied. An air pressure of 100 lb. per sq. in. was available for these tests.

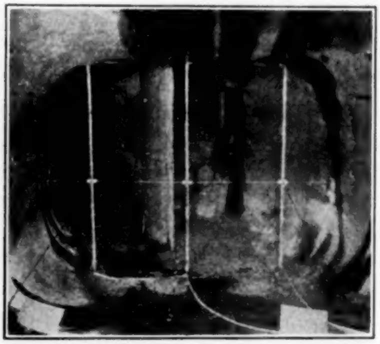


FIG. 21 LOWER ELECTRODES A, B, AND C

79 At first, six pairs of small electrodes attached to the distribution pipes and placed about two inches downstream from the pop valves recorded the introduction of the salt at the upper end of the penstock. These electrodes were made of copper plates and had an area of about two square inches each. They were placed parallel to the thread of the penstock and were numbered from 27 to 32, inclusive. Electrodes Nos. 31 and 32 are shown in the photograph of the distribution pipes and pop valves in bay C. (See Fig. 19.) Later in the field tests these electrodes were moved upstream to the plane of the pop-valve faces and were placed at an angle of 45 deg., but retained the same numbers. Still later, larger electrodes were used in the original locations.

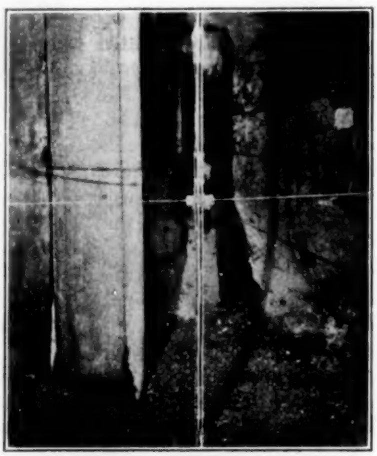


FIG. 22 ELECTRODE B

80 Other methods of recording salt introduction were a contact signal placed on the handle of the quick-acting valve and an electric signal operated by water pressure through a gage connected with the distribution pipe in bay C.

81 During the last period of these tests and during the final efficiency tests of unit No. 7, six pairs of electrodes, placed 22 in. downstream from the plane of the pop valves, were used. These electrodes, Nos. 34–39, made of 4-in. by ½-in. steel plate, 10 ft. long, were placed parallel to the thread of the penstock.

82 Three pairs of steel electrodes, placed vertically at the lower end of the penstock and about three feet upstream from the entrance to the scroll case, recorded the passage of the salt at that section. These lower electrodes were made of 4-in. by $\frac{1}{2}$-in. steel plates spaced 1 in. apart by horn fiber insulation and with tie bolts insulated with fiber and rubber. These plates were fastened to, but insulated from, angle irons bolted to the roof and floor of the penstock. These electrodes were continuous, extending from the floor to the roof. The six plates were numbered from 1 to 6 and the three pairs lettered A, B, and C. (See Figs. 21 and 22.)

83 Two pairs of shorter electrodes made of steel plates 3 ft. long were placed 6 in. from the side walls of the penstock and were used during a portion of the tests. These electrodes were lettered X and Y. These locations are shown in Fig. 29.

84 Other lower electrodes were made by placing 4 x 12 x $\frac{1}{4}$-in. copper strips between the two steel electrodes at 15 points with five on each pair of electrodes, A, B, and C. These copper strips were not in the center of the space, but were $\frac{1}{4}$ in. from one steel and $\frac{1}{2}$ in. from the other steel. They were connected so as to form a pair of electrodes with either or both steel plates, and were numbered from 7 to 21, inclusive. These 15 short electrodes are shown in Figs. 21 and 22.

85 In later tests the fiber insulation was changed to hard rubber and finally to porcelain, and the 15 copper strips were removed, but for the last week of the tests nine of these coppers were replaced. In these last electrodes the small copper plates were placed on the outside of the steel plates instead of between them, as in the early electrodes.

86 Wires with switches connected all of the electrodes to the recording electrical instruments in the gate house. The record was made in three ways: by recording curve-drawing ammeter; by recording curve-drawing wattmeter; and by a special integrating device developed by the engineers of the Laurentide Power Company. A detailed description of this last device will be found in Section VI of this paper. A standard clock was fitted with a magnet and contact point, and was so wired through a relay that the movement of the pendulum recorded seconds on the paper chart. The time record was made by both pen and jump spark.

87 During some of the speed and efficiency runs, an electric revolution counter, connected to the wheel shaft, was located on the testing tables, and through a relay and jump-spark device each revolution was recorded on the same chart. This was checked by an observer, on the counter. Fig. 23 is a wiring diagram, and Fig. 24 is a photograph of the electrical apparatus.

88 Float gages recorded the head- and tail-water elevations. A water column connected to the piezometer plates in the penstock recorded the pressure head at the entrance to the scroll case.

89 During practically all of these tests other records in connec-

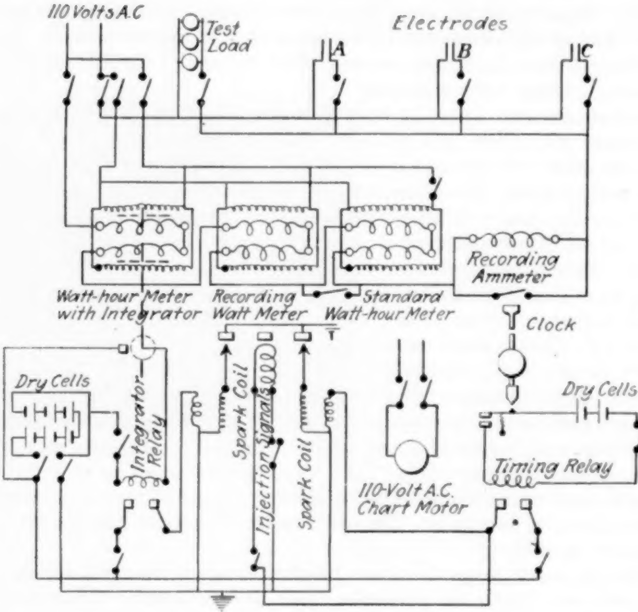


Fig. 23 Wiring Diagram, Laurentide Power Company Tests

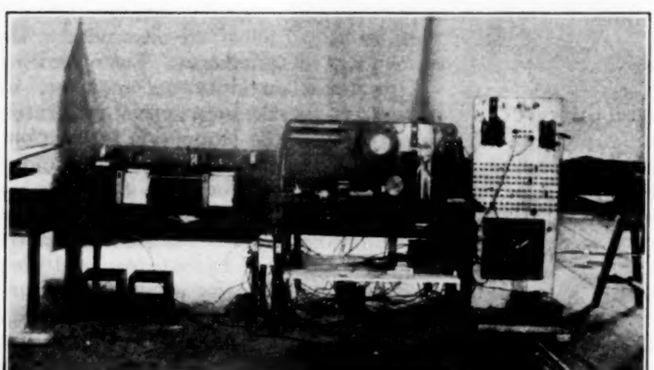


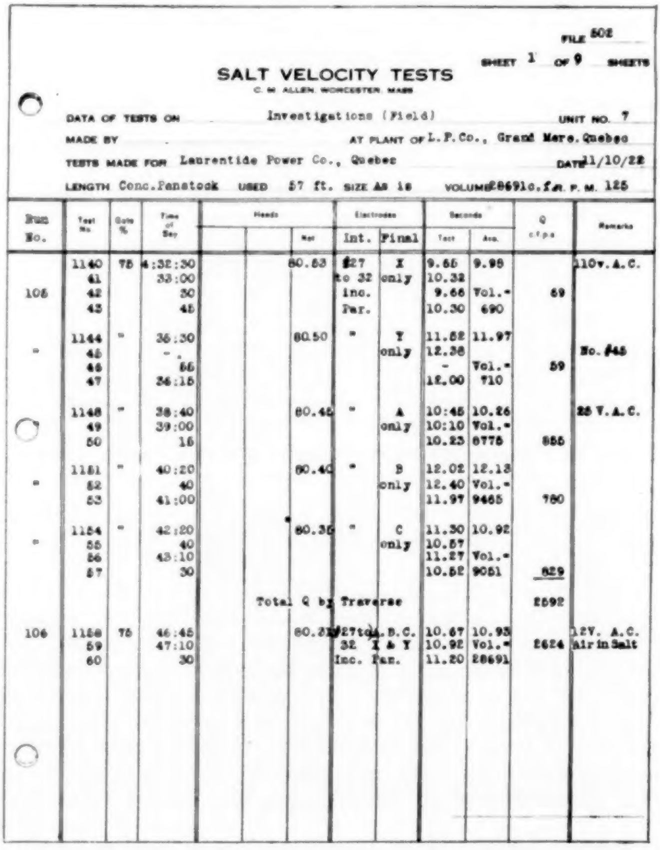
Fig. 24 Electrical Apparatus Used in Laurentide Power Company Tests

tion with the unit, such as load, head, and gate opening, were observed and recorded, and during the latter portion of the tests many of the runs were efficiency runs on the water wheel. The

electrical load was obtained from an integrating watt-hour meter, and checked by indicating wattmeters located in the generator room of the power house.

Description of Tests

90 During all important tests, observers were stationed at the head- and tail-water gages, at the water column recording pressure



SALT VELOCITY TESTS

C. M. ALLEN, WORCESTER, MASS

FILE 502

SHEET 1 OF 9 SHEETS

DATA OF TESTS ON Investigations (Field) UNIT NO. 7

MADE BY — AT PLANT OF L.P.Co., Grand Mere. Quebec

TESTS MADE FOR Laurentide Power Co., Quebec DATE 11/10/22

LENGTH Conc.Penstock USED 57 ft. SIZE AS 18 VOLUME 28691c.f. R.P.M. 125

Run No.	Test No.	Gate %	Time of Day	Heads			Electrodes		Seconds		Q c.f.p.s	Remarks
						Net	Int.	Final	Test	Ave.		
	1140	75	4:32:30			80.53	#27	X	9.65	9.98		110v.A.C.
	41		33:00				to 32	only	10.32			
105	42		30				inc.		9.66	Vol.-	69	
	43		45				Par.		10.30	690		
	1144	"	35:30			80.50	"	Y	11.52	11.97		
"	45		- .					only	12.38			No. #45
	46		55						-	Vol.-	59	
	47		36:15						12.00	710		
	1148	"	38:40			80.45	"	A	10:45	10.26		25 V.A.C.
	49		39:00					only	10:10	Vol.-		
	50		15						10.23	6775	655	
	1151	"	40:20			80.40	"	B	12.02	12.13		
"	52		40					only	12.40	Vol.-		
	53		41:00						11.97	9465	780	
	1154	"	42:20			80.35	"	C	11.30	10.92		
"	55		40					only	10.57			
	56		43:10						11.27	Vol.-		
	57		30						10.52	9051	829	
					Total Q by Traverse						2592	
106	1158	75	46:45			80.31	#27 to	A.B.C.	10.67	10.93		12V. A.C.
	59		47:10				32	X & Y	10.92	Vol.-	2624	Air in Salt
	60		30				Inc.	Par.	11.20	28691		

Fig. 25 Sample Data Sheet of Laurentide Power Company Tests

head, at the electrical instruments in the generator room, at the wheel governor, and in the operating room of the power house. The test was directed from the gate house, where observers on the salt velocity meters were connected by telephone with the observers in the power house.

91 The maintenance crew of the power house provided most of the observers, performed all the labor in connection with the installation of apparatus, and attended to the brine mixing and distribution during the tests.

92 All tests were numbered, and all tests and observations were

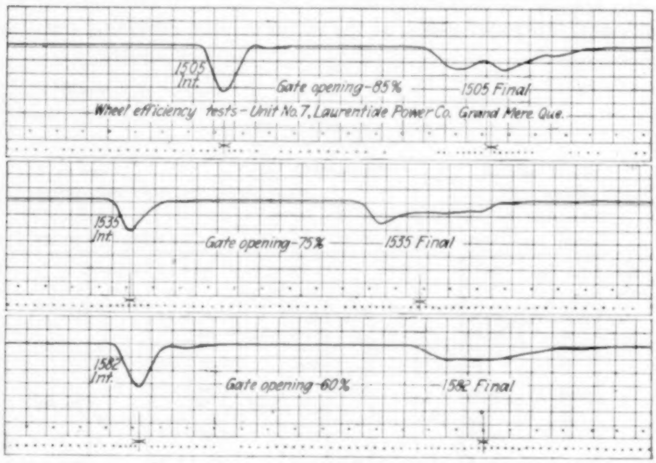


FIG. 26 SAMPLE CURVES, LAURENTIDE POWER COMPANY TESTS

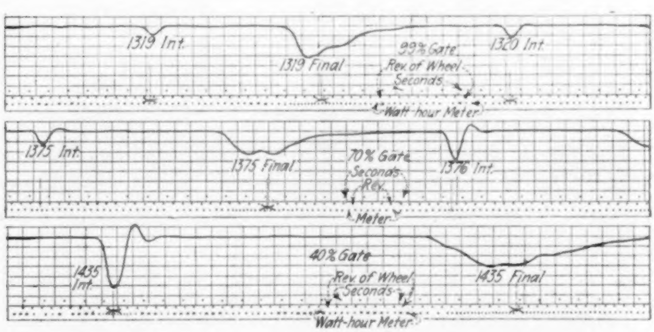


FIG. 27 SAMPLE CURVES, LAURENTIDE POWER COMPANY TESTS

recorded against the time of day. The watches of all observers were synchronized at the beginning of a test, and were checked at the end of the test. The necessary notes and data were recorded on both the moving chart and on the data sheets prepared for the purpose. Fig. 25 is a sample data sheet.

93 Including trials and all efficiency tests, about 1800 individual

tests or charges of salt solution were used. These were grouped into 220 runs of from five to twenty shots each.

94 Figs. 26 and 27 show sample curves recording the passage of salt at both the upper and lower electrodes for six different gate openings. All of these curves were made in the November tests after the various electrodes had been changed and improved.

Computations

95 The dimensions of the penstock were carefully measured by local engineers before the beginning of the tests, and the volumes between the vertical planes of the various electrodes and the pop valves were accurately computed. Then from the charts the number of seconds required for the passage of the salt between two stations was counted. The volume in cubic feet for that section,

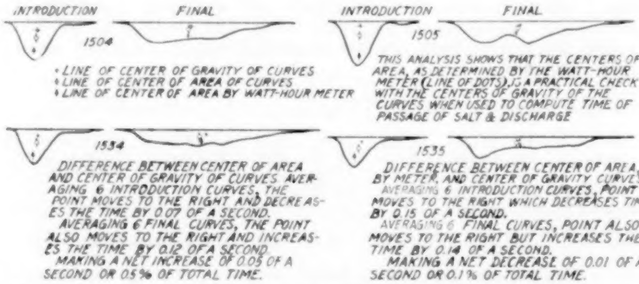


Fig. 28 Analyses of Curves

divided by the time of passage in seconds, equals cubic feet per second.

96 Referring to the sample charts, Figs. 26 and 27, the curves are the pen records of the passage of the brine across the electrodes with current through a wattmeter. The line of dots at the bottom of the charts indicates the passage of the brine across the same electrodes with current through the watt-hour meter. The middle line of dots (November 13 only) indicates the revolutions of the wheel, and the upper line of dots on all charts indicates the time in seconds.

97 As in the laboratory investigations at Worcester, considerable time was given to the problem of what point to use in computing time. Fig. 28 is an analysis of the six sample curves, for both introduction and final passage. The axes of the centers of gravity of the pen curves have been carefully determined by cardboard figures balanced on a knife edge. The axes of the centers of area of the same curves have been accurately fixed by a planimeter, and both of these axes are shown, together with the axes

of the centers of area as determined by the line of dots at the bottom of the sample curves. A study of this analysis shows that the results obtained by taking the centers of area, as indicated by the jump-spark device through the watt-hour meter on the original curves, differs by only one-tenth of one per cent from the results obtained by using the center of gravity of the pen curves made by the wattmeter.

98 Theoretically the center of gravity of the salt charge which is indicated on the chart by the center of gravity of the area enclosed by the curve is the correct point from which to compute time. In a symmetrical curve the maximum deflection, or the peak of the curve, is on the same axis as the center of area and the center of gravity of the curve, and all three will indicate the same time for the passage of the salt. The curves made by these tests

TABLE 5 COMPARISON OF RESULTS BY VARIOUS METERS ON BASIS OF SECONDS INDICATED FOR THE PASSAGE OF SALT FROM INTRODUCTION TO FINAL ELECTRODES. ALL AT 60 PER CENT GATE

Run No.	No. of tests	Meters D	E	F	Remarks
14	13	14.06	14.42	14.31	Salt in at all bays. Final at all bays
14a	8	14.36	14.55	14.50	Salt in at C only. Final at all bays
15	5	14.51	14.82	14.50	Salt in at all bays. Final at C only
Average, 3 runs		14.31	14.60	14.44	
Average, all 9 runs = 14.45					Not weighted for number of tests
Per cent plus or minus..........		−1.0	+1.0	0.0	Avg. of 9 = 100 per cent.

Meters D and E were Prof. Allen's Bristol ammeters. Meter F was Laurentide Power Company's General Electric wattmeter. The "F" meter was apparently more reliable and accurate (as well as more convenient on account of watt-hour meter attachment) and was used for the majority of future tests.

were not as symmetrical as those obtained in the laboratory investigations, and the maximum deflection of the meter, that is, the peaks of the curves, would not give accurate results. But the curves were not distorted sufficiently to cause any material difference between the center of area and the center of gravity.

99 During the majority of the tests the point on the curves from which time was computed was the center of area, as indicated by the watt-hour meter record. In some of the early tests the records of the ammeters were used, and Table 5 shows the comparative results with the various meters. The computations were simpler with the wattmeter and the watt-hour meter device, and the results using the axis of center of area by that method coincided so closely with the center of gravity of the curves that any difference was ignored.

100 Since these investigations were made, the engineers of the Laurentide Power Company have made some tests investigating the proportionality or relation between the density of the salt and the electrical conductivity of the solution. The curves in Fig. 29 show the conductivity in amperes plotted on the density of salt

solution by an arbitrary scale running up to 220. While the density of the salt solution was practically the same for all tests (specific gravity 1.18 to 1.20), the density of the solution when mixed with the penstock water and passing down the penstock changed decidedly. This is shown by the relative length of the introduction and final curves, the final curves being three times as long as the introduction curves. And when the varying velocities of the water are considered, we find that the dilution is at least six times as much, i.e., the density of the solution is only one-sixth as much at the lower end of the penstock as at the upper end. Since all of the solution actually passing the introduction electrodes was never

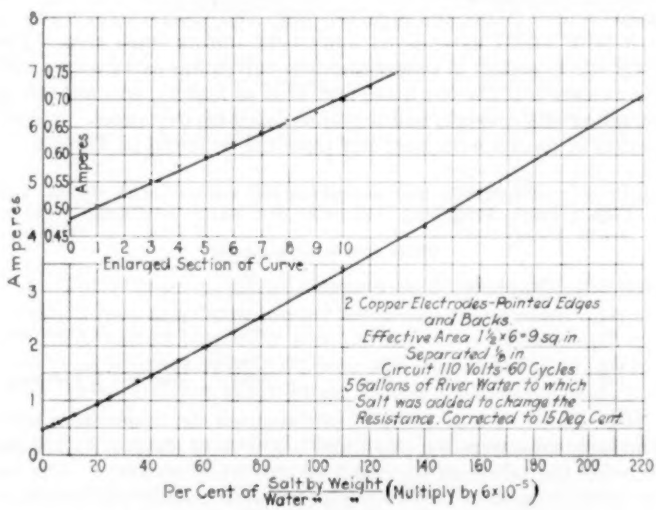


Fig. 29 Curve of Salt Density on Conductivity

over ten on the scale of density, and the curve from zero to ten is practically a straight line, any error caused by varying density of salt solution at the different electrodes can be ignored.

Results

101 Since these tests were mainly to determine the best apparatus and methods to be used in the efficiency tests on unit No. 7, no summary of results has been prepared. Summaries indicating the accuracy of the method are to be found in the sections on laboratory investigations. All of the result sheets shown are comparisons of results using various apparatus and methods.

102 In the summaries of laboratory investigations the accuracy of the salt velocity method in round penstocks has been shown; in other tests its accuracy in rectangular flumes and

conduits has been proved, but its accuracy in a penstock like that of unit No. 7, at Grand Mere, remained to be settled when these investigations were started in October, 1922. Since there was no standard of measurement for the discharge of unit No. 7 to compare the results with, many check runs were made. The best checks were where runs and comparisons were made between different apparatus and different methods, like checking the results with composite curves by a traverse or point measurement.

103 The general plan of these investigations was a cut-and-try process of elimination. The trials and elimination of apparatus are briefly as follows: The mixing and pressure tanks were satisfactory. Increasing the size of the filling pipe of the mixing tank saved time in filling the pressure tank, and twice the capacity of each tank would have saved more time during some runs. The distribution system of piping and pop valves was satisfactory with one exception. The fire hose, first used as leaders, was not strong enough or stiff enough, and it was replaced by iron pipe. A comparison of results with hose and pipe will be found in Table 6.

TABLE 6 COMPARISON OF DISCHARGES, USING FIRE HOSE AND USING IRON PIPE FOR LEADERS ON SALT DISTRIBUTION. AT 80 FT. HEAD

(Other conditions the same)

Run No.	No. of tests	Gate, per cent	Q cu. ft. per sec.	
150	10	80	2783	Hose
186	5	80	2713	Pipe = − 2.5 per cent
158	10	40	1382	Hose
193	5	40	1363	Pipe = − 1.4 per cent

104 Three or four types of introduction electrodes were tried. The early ones were too small and too close to the pop valves and recorded the salt from too small an area to be accurate. The last set of six long introduction electrodes were reliable and accurate. Table 7 is a comparison of results with old and new introduction electrodes. The new electrodes indicate slightly greater discharge. This is accounted for by the theory that some of the fast water went around the old introduction electrodes because they were of small area and too near the pop valves.

TABLE 7 COMPARISONS OF DISCHARGES, USING OLD AND USING NEW INTRODUCTION ELECTRODES

(Iron pipe leaders in each case. At 80 ft. head.)

Run No.	No. of tests	Gate, per cent	Q, cu. ft. per sec.	
186	5	80	2713	Old electrodes
185	5	80	2739	New electrodes = + 0.6%
183	5	40	1362	Old electrodes
193	5	40	1363	New electrodes = + 0.1%

105 Table 8 gives a comparison of results by using various combinations of final electrodes at 49.5 per cent gate opening. This table also shows the results obtained by giving each elec-

trode one-third the volume of the penstock and by an arbitrary computation of the volume for each electrode, based on dividing the penstock by lines from the centers of the piers at the upper end of the penstock to points midway between the final electrodes. No method of accurate determination of these three volumes is known, but it is quite certain that they are not equal.

TABLE 8 COMPARISONS OF DISCHARGES COMPUTED BY USING VARIOUS FINAL ELECTRODES. AT 80 FT. HEAD

Run No.	No. of tests	Gate, per cent	Q, cu. ft. per sec.	Methods of computing	Electrodes
1	6	49.5	1653	Estimated volumes, 3 threads	5 coppers paired with 1 steel at A, B, & C. (Parallel.) 3 curves to compute.
1	6	49.5	1683	Equal volumes, 3 threads	
5, 6, 7	58	49.5	1686	Equal volumes for 15 threads. (Point method.)	1 steel and 1 copper paired at 15 points. 15 curves to compute total.
8	13	49.5	1657	Estimated volumes, 3 threads	2 steels paired (no coppers) at A, B, & C. 3 curves to compute.
8	13	49.5	1649	Equal volumes, 3 threads	
13	7	49.5	1654	Total vol. of penstock used	2 steels paired and 1 steel paired with 5 coppers at A, B, & C all connected in parallel. 1 curve to compute.
Average			1664		

106 The maximum and minimum values for discharge shown in Table 8 vary by about one per cent from the average. Eliminating the values computed with threads of equal volume, the values remaining are 1653, 1654, and 1657, which are close checks.

107 Table 9 shows the volumes of the above threads and of all other volumes used between various stations and electrodes.

TABLE 9 VOLUMES USED FOR PENSTOCK. SALT VELOCITY TESTS ON UNIT NO. 7, OCT. AND NOV., 1922.

(Final electrodes at cross-section C in every case. Various signals for salt introduction.)

Introduction signal	Length horizontal, ft.	Thread	Volume of thread, cu. ft.	Total volume, cu. ft.
Handwheel and gage on pop valves	48.00	A B C	9093 10679 9045	28817
Old introduction electrodes, Nos. 27 to 32 inc.	47.80	A B C X Y	8775 9465 9051 690 710	28691
Using both of above signals. Average of two	47.90	A B C	9021 10662 9071	28754
New introduction electrodes, Nos. 34 to 39 inc.	46.19	A B C	Not computed	27677

108 Table 10 shows comparisons at four different gate openings between results with one composite curve and with three or five

separate curves from the final electrodes *A*, *B*, *C*, *X*, and *Y*; that is, with three or five shots to complete the measurement. These comparisons show very little variation between traverses and

TABLE 10 COMPARISON OF DISCHARGES TAKING FINAL ELECTRODES AS A TRAVERSE, *A*, *B*, *C*, *X*, AND *Y*, SEPARATELY (3 OR 5 CURVES) AND AS A COMPOSITE, 3 OR 5 ELECTRODES CONNECTED IN PARALLEL. (1 CURVE.) AT 80 FT. HEAD

Run No.	No. of tests	Gate, per cent	Q, cu. ft. per sec.			
1	6	49.5	1653	Avg. 1655	Traverse	= 100%
8	13	49.5	1657	*A*, *B* & *C*	Traverse	
13	7	49.5	1654	*A*, *B* & *C*	Composite	= −0.1%
105	25	75	2585	*A*, *B*, *C*, *X*, & *Y*	Traverse	= 100%
106	3	75	2618	*A*, *B*, *C*, *X*, & *Y*	Composite	= +1.3%
187, 188, 189	14	80	2722	*A*, *B*, & *C*	Traverse	= 100%
185	5	80	2730	*A*, *B*, & *C*	Composite	= +0.3%
190, 191, 192	17	40	1370	*A*, *B*, & *C*	Traverse	= 100%
193	5	40	1363	*A*, *B*, & *C*	Composite	= −0.5%
				Average	= Composite	= +0.25%

composites, the average for the four gate openings indicating that the discharge by composite is one-quarter of one per cent in excess of the discharge indicated by the traverse.

109 In connection with these traverses, the direction and veloc-

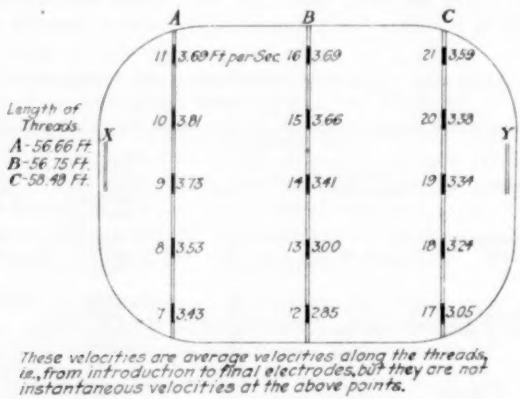


FIG. 30 SKETCH SHOWING VELOCITIES IN PENSTOCK ALONG THE THREADS OF THE 15 COPPER STRIPS USED IN TRAVERSING PENSTOCK AT CROSS-SECTION C

ity of flow in the different parts of the penstock were studied. Fig. 30 shows the location of all electrodes used in making these traverses, together with the velocities in feet per second at each elec-

trode for one gate opening. Fig. 31 shows the curves of these velocities in the penstock. These velocities are not instantaneous velocities, but are average velocities between the introduction electrodes and the final electrodes. These curves are plotted for both horizontal and vertical cross-sections of the penstock and indicate clearly that the fastest water is at the top and at the left, or A, side of the penstock.

110 The dispersion of the salt is indicated in Fig. 32. This shows the results of introducing salt at one bay only and taking the final curve from each electrode separately. Five shots were

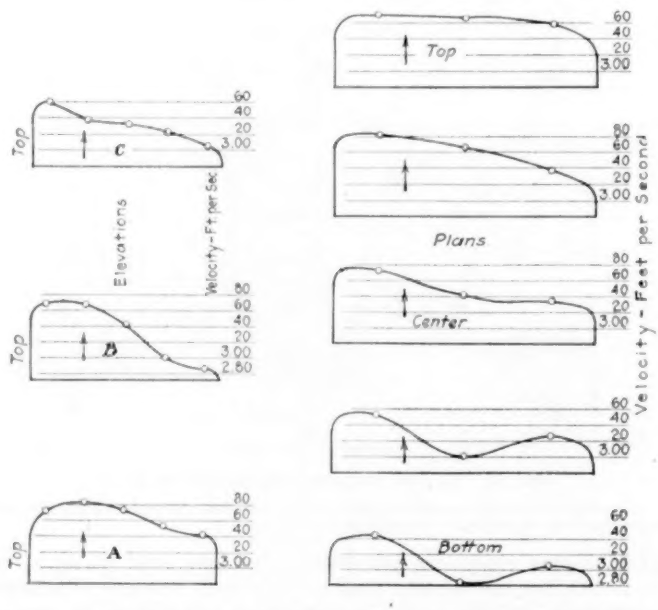


FIG. 31 TRAVERSE CURVES, LAURENTIDE POWER COMPANY TESTS

taken with salt introduced at each bay. The extent of this dispersion or distribution is greater at the smaller gate opening when the velocity of the water is slower. At 50 per cent gate opening, when salt was introduced at B, the center bay, its effect showed at all the final electrodes except Y. This is consistent with the traverse curves showing faster water on the X-side than on the Y-side.

111 As one phase of this investigation, speed runs were made. The generator load was delivered to electric boilers, making a wide range of speed possible. The curves of discharge on r.p.m. for various gate openings are shown in Fig. 33. These speed curves are very consistent with the speed curves derived from the

Holyoke tests of the I. P. Morris Company's model runner. It was possible to pick up slight differences in discharge on change of speed, these differences being as low as three second-feet with total discharges running up to 3000 sec-ft. The bumps in the curves at 65 per cent and 75 per cent gate opening are duplicates of those shown by the Holyoke tests. The results of these speed runs gave added confidence to this method of water measurement, and the resulting speed curves were used in reducing values for the efficiency report on unit No. 7.

112 Early in October the velocity of the water near the roof and floor walls of the penstock was determined by the fifteen

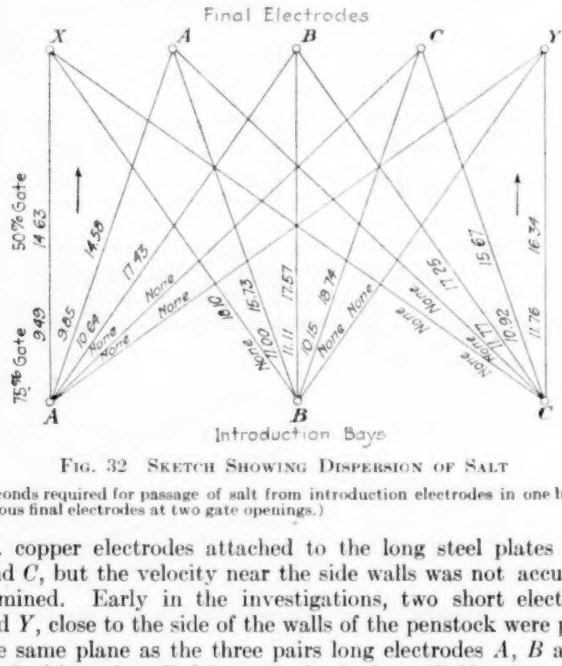


FIG. 32 SKETCH SHOWING DISPERSION OF SALT

(Seconds required for passage of salt from introduction electrodes in one bay only to various final electrodes at two gate openings.)

12-in. copper electrodes attached to the long steel plates at A, B, and C, but the velocity near the side walls was not accurately determined. Early in the investigations, two short electrodes, X and Y, close to the side of the walls of the penstock were placed in the same plane as the three pairs long electrodes A, B and C, which had been installed from the beginning. Table 11 compares tests with and without X and Y.

TABLE 11 COMPARISONS SHOWING EFFECT OF NEW SIDE ELECTRODES *X* AND *Y* USED WITH ELECTRODES *A*, *B*, AND *C*, AT 80 FT. HEAD

Run No.	No. of tests	Gate, per cent	Q, cu. ft. per sec.	—Final Electrodes—	
26	10	75	2643	*A*, *B*, *C*, *X*, and *Y* parallel	
101	24	"	2647	" " " " " " "	
103	3	"	2631	" " " " " " "	
106	3	"	2620	" " " " " " "	
107	5	"	2639	" " " " " " "	
Average			2642	" " " " " " "	Weighted for shots
108	5	"	2634	*A*, *B*, and *C* only parallel	−0.3%

113 These comparisons indicate that the adding of X and Y did not affect the indicated discharge because the velocity at X was only slightly faster than the velocity at A, while the velocity at Y was slightly lower than that at C. The average of five runs

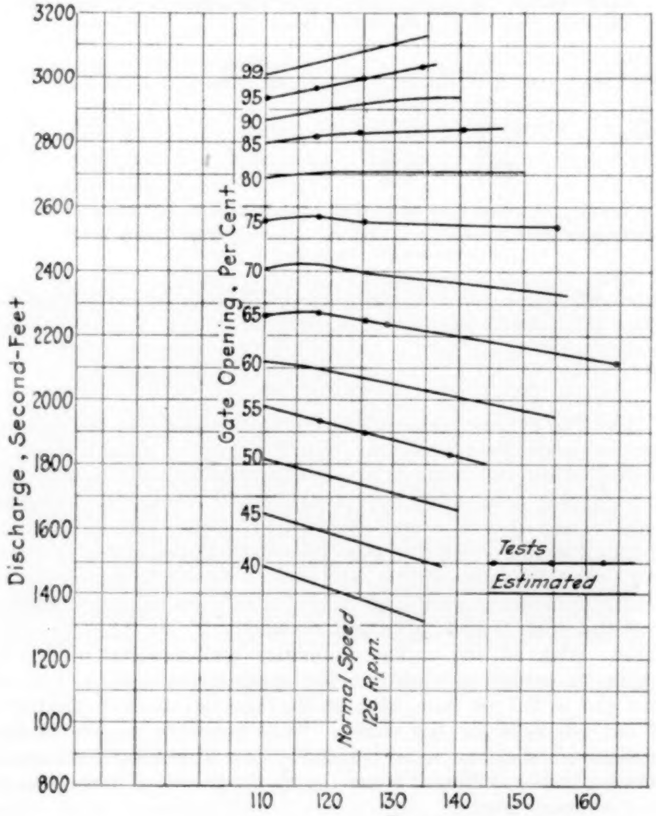


FIG. 33 CURVES OF DISCHARGE AT VARIOUS SPEEDS AT 80 FT. HEAD

shows 2642 sec-ft. with the five electrodes and 2634 sec-ft. with three electrodes.

114 Table 12 is a comparison of the time required for the passage of salt from the introduction electrodes to X and to A, as well as to Y and to C. The time to X was 1.2 per cent faster than the time to A, the nearest long electrode, while the time to Y was 3 per cent slower than the time to C. Averaging, we find that the time to X and Y together was less than 1 per cent slower than to A and C together. Since the areas and the volumes assigned

to the electrodes A and C are twelve times the areas and volumes assigned to X and Y, the effect of X and Y on the total discharge computed from either traverse or a composite curve is less than

TABLE 12 COMPARISON OF TIME REQUIRED FOR PASSAGE OF SALT FROM INTRODUCTION ELECTRODES TO FINAL ELECTRODES A AND C AND TO X AND Y

Gate, per cent	Seconds to X	Seconds to A	Gate, per cent	Seconds to Y	Seconds to C
50	14.63	14.58	50	16.34	15.67
60	13.20	13.55	60	15.80	15.61
65	11.09	11.63	65	13.80	13.11
67.5	11.64	11.02	67.5	13.03	13.03
75	9.98	10.26	75	11.97	10.92
80	9.49	9.85	80	11.76	10.92
Totals........	70.03	70.89	Totals........	82.70	79.26

Seconds for X = − 1.2% Seconds for Y = + 3.0%

Average seconds for X and Y = + 0.9%

one-tenth of one per cent, which checks the comparison in Table 8. After this was determined the short electrodes X and Y were abandoned.

Conclusions

115 The greatest value of these investigations was the demonstration, under field conditions, that the tests by the salt velocity method as applied to the setting at Grand Mere could be repeated and checked indefinitely, and that the tests could be repeated with varying apparatus and equipment and with various methods of computation, and still check. These tests tried out and eliminated several sources of error, and showed that the final apparatus and methods used in the efficiency tests were an improvement over the original apparatus first installed.

116 With all the various apparatus and methods used, the maximum variations shown in the comparisons were + 1.3 per cent and − 2.5 per cent, with an average variation of one-tenth of one per cent for the whole. With improved apparatus the maximum variations were reduced. For all comparisons made between tests at different times the discharge values were reduced to a common head.

V — 1922 COMMERCIAL TESTS

117 During the fall of 1922, ten successful commercial tests were made using the salt velocity method. On all of the tests the object was the measurement of the discharge through the penstock of a power plant. On seven of the tests the discharge values were used in computing the efficiency of the units, on two tests the discharge values were used to calibrate meters, and in the tenth test the values were used for determining the efficiency of the unit and also for calibrating a Johnson valve to be used as a venturi meter.

Apparatus and Methods

118 The plants under test were all hydroelectric power plants. The sizes of penstocks used were:

Two tests — Riveted steel	11 ft. diameter,	1000 ft.	long.
Two tests — " "	12 " "	500 "	"
Two tests — " "	13 " "	1500 "	"
One test — Reinforced concrete	20 " "	500 "	"
Three tests — Concrete.	Rectangular tapered, 10 to 70 ft. long.		

119 The tanks used varied from 150 to 1200 gal. in capacity and the salt pipes from 2 to 4 in. in diameter. Fig. 34 shows one of the large installations.

120 *Salt Introduction.* In all cases the salt solution was controlled by a quick-acting valve in the supply pipe leading from

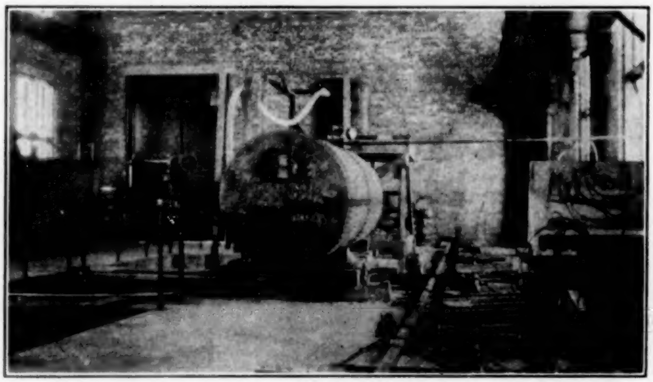


Fig. 34 Salt Tanks and Distribution Pipes for Test of 50,000-hp. Unit

the pressure tank, and was introduced into the penstock through pop valves. In the steel pipes one $2\frac{1}{2}$-in. pop placed at the center of the penstock entrance was used. In the concrete penstocks multiple pops of various sizes were used, as many as 24 being installed for one penstock.

121 *Electrodes.* The electrodes were of various forms and sizes, and both steel and copper were used. For insulation of these electrodes horn and fiber, hard and soft rubber, and porcelain were used, and on the later forms waterproof shellac was freely used on all contact surfaces. The spacings between pairs of electrodes varied from $\frac{1}{16}$ in. to 4 in. In most cases two plates were used for each set, but in the others three parallel plates were used to get more surface and a greater deflection of the meter needle. Fig. 35 shows a design using three copper plates.

122 For the circular penstocks one set of electrodes was used

for finals at the lower end, but on the rectangular penstocks multiple sets were used to cover the cross-section. In one case a complicated system of electrodes was used. By a combination

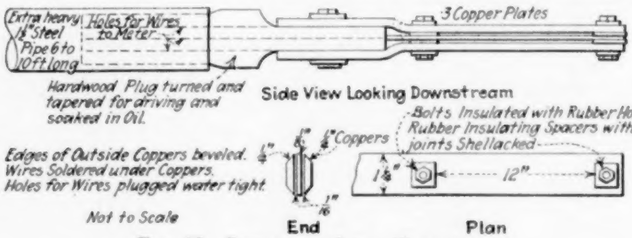


FIG. 35 SKETCH OF 3-PLY ELECTRODE

of long steel and short copper plates 35 electrodes were placed in one plant to cover a cross-section of 335 sq. ft. area. Figs. 36 and 37 show two forms of electrodes used on a test of unit No. 6

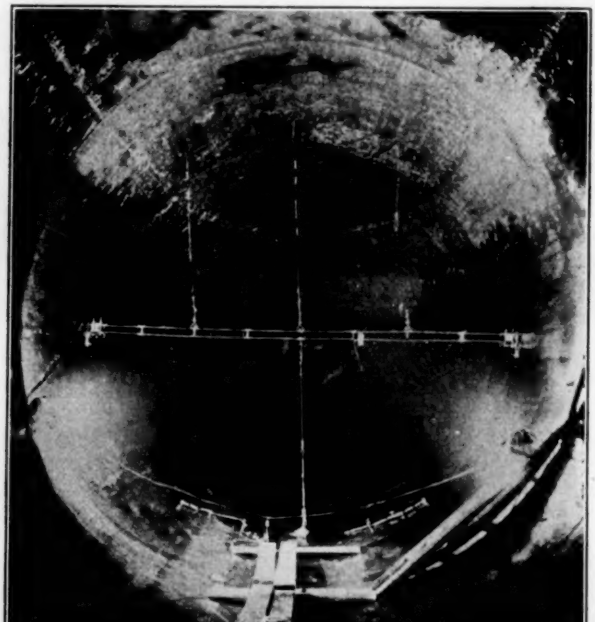


FIG. 36 ELECTRODE USED IN CONCRETE PENSTOCK 20 FT. IN DIAMETER

for the Shawinigan Water and Power Company, at Shawinigan Falls, Quebec.

123 For recording the introduction of salt at the upper end of the penstock, both switches and electrodes were used. On

all the long pipes a contact switch was installed on the handle of the quick-acting valve, but on all the short penstocks upper electrodes were placed close to the pop valves. In several cases the upper electrodes were fastened to the pop valves.

124 *Meters.* All introduction signals and all electrodes were wired to the recording meters. These meters were the same instruments as were used in the 1922 field investigations. Alternating current at 110 volts was used in these circuits, but at times portable transformers were used to change the voltage and current

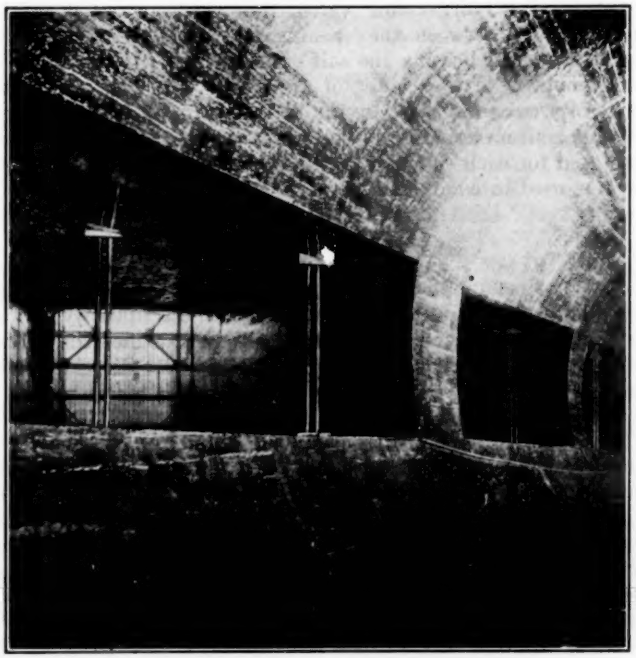


FIG. 37 ELECTRODES USED IN GATHERING TUBES

in regulating the deflection of the meter needle and the height of the curves.

125 *Timing.* On some of these tests a standard seconds-pendulum clock was wired to the recording meters, and by means of batteries, a relay, and a spark coil, jump-spark dots recorded seconds on the chart. For the remainder of these tests no clock was available and a seconds pendulum with a magnetic weight was used. This pendulum was made in the field, and during the first test did not beat exact seconds, making a time calibration necessary. For all the later tests the pendulum was adjusted to beat exact seconds and was frequently checked.

126 *Standards of Measurement.* No standards of water measurement were employed on these tests and no other methods used except when a meter was being calibrated. At one plant the same unit was tested by the Gibson method, and the final curves by the two methods checked exactly along the range of high efficiency and only varied slightly at the lower gate openings.

127 *Computations.* Prior to a test the volume of a penstock was always accurately determined from plans and checked by field survey.

128 When the salt introduction was recorded by a signal switch on the quick-acting valve, time was computed from a point midway between the opening and closing of that valve. For all curves indicating the salt passing the electrodes, the time was computed to the center of gravity of the curve or to the center of area as indicated by the line of jump-spark dots through the watt-hour meter. From 5 to 10 shots or charges of salt solution were used for each run, and the average of all the shots during a run was used in computing the discharge.

Results and Conclusions

129 The results of the tests on the long, round penstock were consistent and confirmed the accuracy and reliability of the method applied to such penstocks. They also justified the various apparatuses and methods of computation used on the rectangular, tapering penstocks, but they did not shed any additional light on the accuracy of the method as applied to that type of penstock.

130 However, one of these tests on a large unit in Canada did confirm the accuracy of the salt velocity method on rectangular converging tubes. The penstock for this unit was 500 ft. long with a uniform diameter of 20 ft., and was fed by four rectangular converging tubes and one elliptical diverging tube. During these tests it was possible, at five different gate openings, to compare the discharge measured in these five tubes with the same discharge measured in the main penstock of uniform cross-section. The greatest variation was 0.7 per cent, and the average of all discharges checked exactly. These four rectangular converging tubes were typical of the penstocks at Grand Mere, and this checking of results gave added assurance to the accuracy of the tests on rectangular tapering penstocks.

VI — 1923 LABORATORY INVESTIGATIONS

131 Following the field investigations at Grand Mere, Quebec, which were completed in November 1922, another set of laboratory investigations was made at Worcester, in January, February, and March, 1923. The main object of these investigations was to determine the degree of accuracy of the salt velocity method against the weighing tank, and particularly the accuracy of the

apparatus and methods of computation used during the field and efficiency tests of 1922.

Apparatus and Methods

132 *Plant.* An 8-in. steel pipe, reduced to 6 in. and 4 in., and branching from the main penstock of the Alden Hydraulic

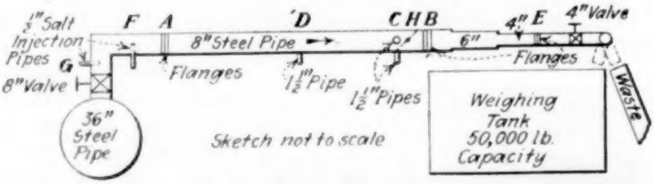


Fig. 38 Sketch of 8-in. Steel Pipe, Worcester Polytechnic Institute Laboratory

Laboratory, was used for these later investigations. Fig. 38 shows this pipe and the stations used in these tests.

133 *Apparatus.* The brine was mixed in a 50-gal. open mixing tank, and a hand force pump was used for the salt injection through $\frac{1}{2}$-in. pipes. These pipes entered the 8-in. pipe at two

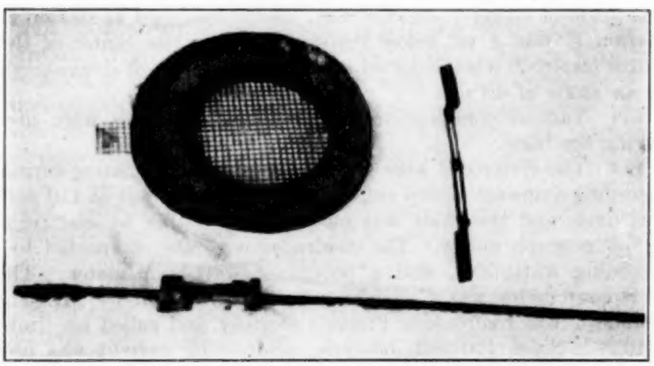


Fig. 39 Electrodes Used in Laboratory, 1923

stations, F and G. F was 3 ft. downstream from a 90 deg. bend in the 8-in pipe, and G was 1 ft. upstream from the same bend. Both of these injection pipes were controlled by valves, and pop valves, open ends, and perforated pipes were used for the salt distribution.

134 Various electrodes were used. Two wire screens ($\frac{1}{4}$-in. mesh), spaced from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. apart, were used at stations A, B, and E. A traversing electrode was made of $\frac{3}{4}$-in. by $\frac{1}{16}$-in. copper plates, with $\frac{3}{8}$ in. of the plates exposed. Four of these plates,

spaced $\frac{1}{16}$ in. apart, were placed parallel to each other. This electrode attached to a pitometer rod was used at station C.

135 The so-called "flat" electrodes were made of two $\frac{3}{4}$-in. by $\frac{1}{16}$-in. copper plates 8 in. long, placed parallel and spaced $\frac{1}{8}$ in. apart. These electrodes were used at stations A, C, and D. When used at station A these plates were fastened between the pipe flanges, and when used at C and D they were fastened to pitometer rods and inserted through stuffing boxes.

136 The so-called "improved" electrodes were made of the same plates as the flat electrodes, but with different spacing. The $\frac{1}{8}$-in. spaces at the ends remained unchanged, but the center spacing was increased to $\frac{1}{2}$ in. and to $\frac{3}{4}$ in. These electrodes were used at stations A, C, and D. Photographs of four electrodes are shown in Figs. 39 and 40.

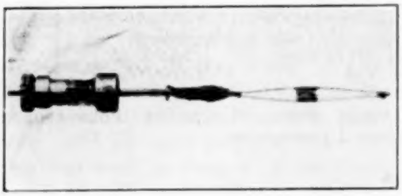


FIG. 40 IMPROVED ELECTRODE USED IN LABORATORY, 1923

137 In order to simulate the conditions in the unit No. 7 penstock at Grand Mere, where the perpendicular final electrodes were at an angle with the direction of the water flow, one improved electrode was made of longer plates (11 in.), which was used at station H. Station H was 4 in. below station C and is the center of this longer electrode when inserted at station C and bent downstream at an angle of 45 deg.

138 Various combinations of all these electrodes were used during the tests.

139 The electrodes were wired to a Bristol alternating-current recording ammeter of five amperes capacity. Current at 110 volts was used, and the chart was motor-driven, either by electric or by phonograph motor. The electrodes were also connected to a recording wattmeter, and a polyphase watt-hour meter. This watt-hour meter was a part of a device developed by Mr. Thibeault of the Laurentide Power Company and called an "integrator." Since 110-volt, 60-cycle, alternating current was used in the salt electrode circuits, it was necessary to use alternating-current instruments. The wattmeter and the watt-hour meter were of the commercial polyphase type (although single-phase current was used) with the two potential elements in parallel and the two current elements in series to double the torque and augment the quickness of the instruments to respond to changes in the load. The recording ammeter and recording wattmeter were of the ordinary type and need no special mention, but the action of the integrator requires detailed description.

140 The object of this integrator is to determine by exact, simple, easily workable, and graphical means the instant when

one-half of the salt-solution charge has passed the electrode. A commercial watt-hour meter is connected with its current coil in series with the electrodes and with its potential coil energized by a constant voltage. On the rotating shaft of the meter is mounted a simple circuit interrupter giving 24 interruptions per revolution to its circuit, which is a local direct-current circuit through the actuating coil of a simple contact-*opening* relay. These contacts in turn energize the low-tension side of a spark coil of which the high-tension leads are connected to a spark gap. A spark punctures the recording-wattmeter chart at every interruption of the interrupter circuit.

141 The spacing between punctures will vary inversely as the speed of the watt-hour meter disk, and consequently inversely as the conductivity of the liquid between the electrodes. While water only is passing the spacing should be uniform; but while the salt solution charge is passing the spacings become smaller. This corresponds to the curve of the recording instrument, the ordinates of which are low and uniform under the former condition and higher under the latter.

142 The recording wattmeter gives a measure of the amount of salt in the charge (assuming constant velocity and a straight-line relationship between salt density and conductivity) in the area of that part of the curve lying above the horizontal line representing the conditions when no salt is passing.

143 A measure of the salt passing is also recorded by the integrator. The horizontal line on the chart, indicating a uniform condition of conductivity of normal water, is here replaced by its exact equivalent in a series of uniformly spaced punctures. Conductivity in excess of this value occurring during the passage of salt demonstrates itself proportionally by the increase in frequency of the punctures.

144 Referring to a chart containing a complete record of the passage of a salt charge by the integrator, Fig. 26 or 27, the time when the solution started to pass and the time it left the electrodes are noted by observing when the spacing of the lower line of punctures changes. Measuring the distance between these two (initial and final) points and dividing by the length of the normal spacing determines how many punctures would have occurred if no salt had passed. By counting the actual number of punctures occurring during the period of salt passage just selected and subtracting from this number the number derived as above, representing the number of punctures due under normal conditions of flow of water with no salt added, the number of excess punctures is determined.

145 The resultant figure of excess is thus proportional to the amount of salt in the charge. One-half this figure of excess is also proportional to one-half the quantity of salt in the charge. Therefore, if that point on the chart is found where the actual number of puncture spacings minus the number due to the water before

dosing with salt, equals one-half the excess of the total passage, the moment is established when one-half of the salt charge has passed. To facilitate the determination on the chart of this point, a simple portable scale was adopted which greatly reduced the amount of work required.

146 To provide a time scale on the charts of the recorders of all types, the same device of puncturing the sheet with a jump spark was employed, the spark being controlled by a relay actuated in turn by a contact-making clock that marks one-second intervals on the chart. A wiring diagram is shown in Fig. 23.

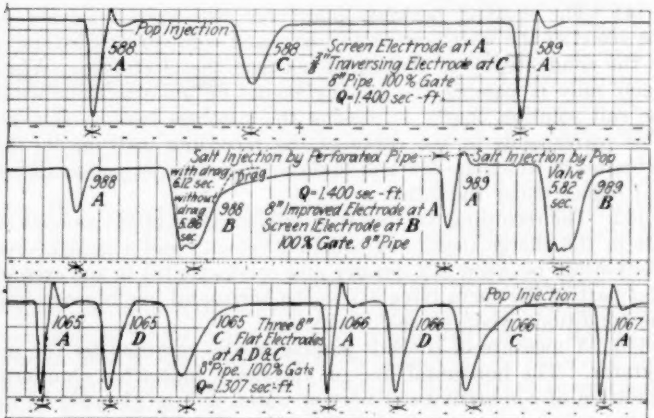


FIG. 41 SAMPLE CURVES OF TESTS ON 8-IN. STEEL PIPE

147 The standard for the water measurement during these tests was a copper-lined weighing tank of 50,000 lb. capacity. The scales were frequently checked with standard weights and were very accurate. They were also very sensitive, a difference of ¼ lb. showing on the balance arm.

DESCRIPTION OF TESTS

148 All of the apparatus and all observers for these tests were in one room. One observer operated the salt pump. Another operated the switches and the recording meters and directed the tests. A third man operated the weighing tank and kept the data noted. During a part of the time a fourth observer was used in computing curves and changing pipe and electrode connections.

149 All tests were numbered and recorded against the time of day, and the necessary notes were kept on both the meter charts and the data sheets. Sample curves are shown in Figs. 41 and 42.

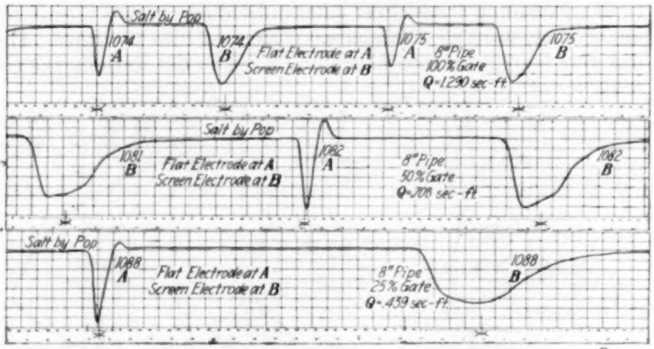


FIG. 42 SAMPLE CURVES OF TESTS ON 8-IN. STEEL PIPE

FILE 503

SHEET 1 OF 14 SHEETS

SALT VELOCITY TESTS

C. M. ALLEN, WORCESTER, MASS.

DATA OF TESTS ON Investigations (1923) UNIT NO. -

MADE BY AT PLANT OF W. P. I. LAB. Worcester

TESTS MADE FOR Laurentide Power Co., Ltd. Quebec DATE 2/4/23

LENGTH Steel Pipe USED A-D 9.77, A-C 20.21, A-B 22.65 SIZE 8"+ VOLUME 3.526, 7.548, 8.130 R. P. M. -

Run	Test No.	Gate %	Time of Day P.M.	[illegible]			Electrodes		Seconds		Q c.f.p.s.	Remarks
				Wt.	Lbs. Sec.	Wt. c.f.	Int.	Final	Test	Ave.		
	1054		2:34	862					D	C		
	1064	100					Flat	2 Flats	2.63	5.65	A-D	3 Flats
	5		6	29388	4893	1.307	A	D & C	2.57	5.57	1.367	A,D & C
128	6						Vol.	Vol.	2.62	5.60	+4.6%	
	7						D=	C=	2.50	5.58	A-C	Pop
	1068		2:40	30217			3.526	7.548	2.58	5.65	1.348	
								Ave.	2.58	5.61	+2.9%	
									B			
	1069	100	2:48	1589			Flat	Screen	6.42			Flat &
	70						A	B	6:17			Screen
	1								6.22			A & B
	2							Vol.=	6.32			Pop
129	3		4	19533	4683	1.304		8.130	6.26	6.26	1.299	
	4								6.27			
	5								6.27		-0.4%	
	6								6.18			
	7								6.24			
	1078		2:52	21122					6.28			
	1079	50	2:56	21113			Same		11.52			"
	80								11.47			
130	1		5	13259	2652	.708	V.=	8.130	11.47	11.51	.707	
	2								11.49			
	1083		3:01	34372					11.59		-0.1%	
	1084	25	3:08	1232			Same		18.75			"
	5								18.38			
131	6					.439	V.=	8.130	18.30	18.44	.441	
	7								18.38		+0.5%	
	1088								18.39			
			14	22968	1641				D	C		
	1089	25					Flat	Flats	8.18	16.50	A-D	3 Flats
	90						A	D & C	7.58	16.57	.484	A,D & C
132	1					.439	Vol.	Vol.	7.70	16.30	+3.4%	
	2						D =	C =	7.79	16.75	A-C	Pop
	3		•				3.526	7.548	7.65	16.77	.455	
	1103		3:22	24200				Ave.	7.77	16.58	+3.6%	

FIG. 43 SAMPLE DATA SHEET OF 1923 LABORATORY INVESTIGATIONS

150 Including trials, over 1300 individual charges of salt solution or shots were used, which were grouped into 161 runs. These runs were segregated into 45 groups, based on the stations the com-

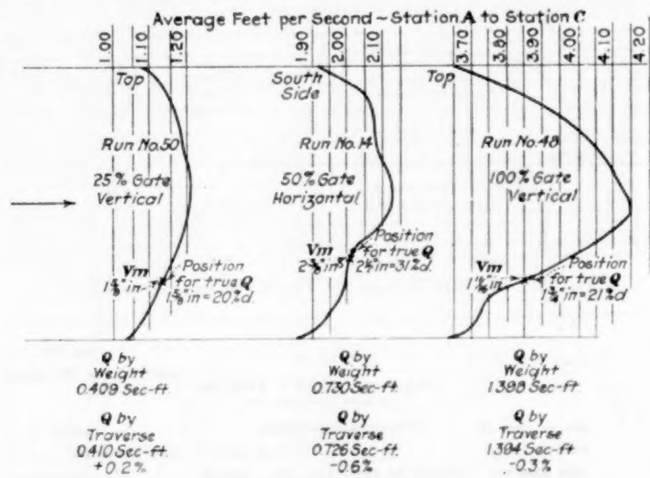


Fig. 44 Traverse Curves for 8-in. Steel Pipe
(Electrodes — 4 plates, 1 × $\frac{1}{8}$-in. copper.)

binations of electrodes, and on the method of computing the curves which were used.

151 At the upper end of the 8-in. pipe, near its junction with the main penstock, was an 8-in. valve which was kept wide open

TABLE 13 PIPE LENGTHS AND VOLUMES AND ELECTRODES USED IN 1923 LABORATORY TESTS

Stations	Electrodes		Length, feet	Volume cu. ft.
A to B	Screen at A	Screen at B	22.57	8.142
A to B	Flat (or Imp.) at A	Screen at B	22.53	8.130
A to C	Flat at A	Flat at C	20.91	7.548
A to C	Screen at A	Flat at C	20.95	7.560
A to D	Screen at A	Flat at D	9.81	3.540
A to D	Flat at A	Flat at D	9.77	3.526
A to E	Screen at A	Screen at E	30.20	9.913
A to E	Flat at A	Screen at E	30.16	9.901
A to H	Screen at A	Flat (45°) at H	21.27	7.678
D to B	Screen at B	Flat at D	12.76	4.602
D to C	Flat at D	Flat at C	11.14	4.022
D to E	Flat at D	Screen at E	20.39	6.377
F to B	Pop at F	Screen at B	23.90	8.626
F to C	Pop at F	Flat at C	22.28	8.040

Area of 8-in. pipe = 0.361 sq. ft.
Volumes determined by weight of water contents. Checked by tape.

during the tests. The gate opening was controlled by a 4-in. valve over the weighing tank at the lower end of the pipe line. This valve was always set at least two minutes before a test to insure a uniform rate of flow, and for the same reason the swivel pipe used to turn the flow in or out of the tank was at the same

angle with the perpendicular at each position and was locked in place. As a rule, all runs were repeated at three gate openings, quarter, half, and full.

152 When starting a test run, the weighing tank was emptied, the waste valve closed, and the tank was carefully weighed. On the next even minute, the operator turned the water into the tank and a few seconds later the recording meters were started and the first shot of salt was injected. Usually from five to ten

TABLE 14

Length of steel pipe used, 22.57 ft. Size, 8 in. Volume, 8.142 cu. ft.

Run No.	No of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Two screen electrodes at Station A and Station B.									
1a	7	100	1.411	1.408	−0.2		Perforated pipe	Peaks of curves or maximum deflection of meter	1
2a	18	100	1.411	1.389	−1.7				
3a	12	70	0.970	0.955	−1.5				
4a	9	25	0.391	0.387	−1.0				
5a	9	25	0.391	0.389	−0.5				
	55					−1.2			
1a	7	100	1.411	1.408	−0.2		Perforated pipe	Center of gravity of curves (by eye)	2
2a	18	100	1.411	1.399	−0.9				
3a	12	70	0.970	0.973	+0.3				
4a	9	25	0.391	0.389	−0.5				
5a	9	25	0.391	0.394	+0.8				
	55					−0.2			
143	10	100	1.356	1.376	+1.5		Pop valve	Center of gravity (by eye)	44
144	10	100	1.311	1.322	+0.9				
145	10	50	0.709	0.702	−1.0				
146	10	25	0.442	0.449	+1.6				
147	10	25	0.443	0.445	+0.5				
148	10	50	0.709	0.701	−1.1				
	60					+0.4			
Screen at Station A. 8-in. improved electrode bent downstream at angle of 45 deg. at Station C. (H) Length pipe used, 21.27 ft. − 8 in. + *d*. Volume, 7.678 cu. ft.									
149	10	50	0.714	0.718	+0.6		Pop valve	Center of gravity (by eye)	45
150	10	50	0.714	0.712	−0.3				
151	10	25	0.396	0.398	+0.5				
152	10	25	0.398	0.397	+0.3				
153	10	100	1.326	1.322	−0.3				
154	10	100	1.322	1.318	−0.3				
	60					+0.1			

Average of 3 groups (omitting group 1 on account of peaks) = + 0.11 per cent.

shots constituted a run, but for the traverses as many as forty shots were made for one run. On the next even minute following the last shot of the run, the tank operator turned the flow of water out of the tank into the waste pipe and again the tank was carefully weighed. The intervals between the two weighings varied from three to twenty-five minutes.

Computations

153 The length and diameter of the various sections of pipe used were carefully measured and the volumes between the stations computed, but these figures were used only as preliminary

TABLE 15

Length of steel pipe used, 22.57 ft. Size, 8 in. Volume, 8.142 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Two screen electrodes at Station A and Station B.									
2	12	100	1.405	1.382	−1.7		Perforated pipe	Center of area by dots including all drag	3
3	8	50	0.693	0.682	−1.6				
4	6	25	0.461	0.458	−0.7				
	26					1.4			
2	12	100	1.405	1.404	−0.1		Perforated pipe	Center of area without drag	4
3	8	50	0.693	0.690	−0.4				
4	6	25	0.461	0.461	0.0				
	26					−0.2			
5	8	100	1.385	1.396	+0.8		Pop valve	Center of area by watt-hour meter (dots)	5
6	6	50	0.773	0.770	−0.4				
8	8	25	0.456	0.456	0.0				
9	13	100	1.396	1.393	−0.2				
10	6	66	0.886	0.884	−0.2				
11	6	50	0.708	0.710	+0.3				
12	5	25	0.410	0.413	+0.7				
	52				+0.1				
8	8	25	0.456	0.451	−0.9		Pop valve	Center of area by watt-meter curves (eye)	6
9	13	100	1.396	1.408	+0.9				
10	6	66	0.886	0.893	+0.8				
11	6	50	0.708	0.712	+0.6				
12	5	25	0.410	0.414	+1.0				
	38					+0.5			
23	7	100	1.410	1.414	+0.3		Open pipe at Station F	Center of area without drag (dots)	7
24	4	100	1.410	1.421	+0.8				
25	4	50	0.736	0.734	−0.3				
26	4	50	0.736	0.739	+0.4				
27	4	25	0.445	0.443	−0.5				
28	4	25	0.445	0.445	0.0				
	27					+0.1			

Average of 4 groups (omitting group 3 on account of drag) = + 0.15 per cent.

TABLE 16

Length of steel pipe used, 22.57 ft. Size, 8 in. Volume, 8.142 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Two screen electrodes at Station A and Station B.									
38	6	100	1.409	1.397	−0.9		Open pipe at Station G	Center of area without drag	8
39	7	50	0.727	0.725	−0.3				
40	5	25	0.431	0.431	0.0				
	18					−0.4			
41	5	100	1.404	1.399	−0.4		Perforated pipe at Station G	Center of area without drag	9
42	5	100	1.406	1.405	−0.1				
	10					−0.3			
86	5	50	0.729	0.727	−0.3		Pop valve	Center of area	10
87	5	100	1.370	1.375	+0.4				
90	6	100	1.372	1.370	−0.1				
	16					0.0			
92	5	100	1.373	1.375	+0.2		Pop valve	Center of area	11
94	7	50	0.719	0.716	−0.4				
95	5	25	0.427	0.428	+0.2				
	17					0.0			
141	6	100	1.317	1.309	−0.6		Pop valve	Center of area	12
142	2	25	0.427	0.427	0.0				
	8					−0.3			

Average of 5 groups = − 0.18 per cent.
Average of 3 sheets (Nos. 2, 3, and 4) all with two screen electrodes at A and B (12 groups, 51 runs and 387 shots), = + 0.07 per cent.

values. A more accurate method of determining the volume, that is, by weighing the same volume of water, was used for the final computations. The section of the pipe was blanked off except for vent holes and filled with water. This water was carefully drawn off into a tank and weighed on calibrated scales. Temperature corrections were made, and the operation repeated and checked.

TABLE 17

Length of steel pipe used, 20.95 ft. Size, 8 in. Volume, 7.560 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
		Screen at Station A. 8-in. improved electrode at Station C.							
20	6	50	0.622	0.627	+0.8		Pop valve	Center of Area	13
21	10	100	1.397	1.410	+0.9				
	16					+0.9			
31	5	100	1.402	1.418	+1.1		Open pipe	Center of area without drag	14
33	5	50	0.740	0.741	+0.1				
34	6	25	0.449	0.452	+0.7				
	16					+0.6			
35	5	25	0.449	0.451	+0.4		Perforated pipe above 8-in. ell.	Center of area without drag	15
36	8	50	0.738	0.741	+0.4				
37	7	100	1.407	1.404	−0.2				
	20					+0.2			
44	6	100	1.407	1.415	+0.6		Perforated pipe below 8-in. ell.	Center of area without drag	16
45	7	50	0.713	0.718	+0.7				
46	5	25	0.425	0.429	+0.9				
	18					+0.7			
85	6	50	0.729	0.727	−0.3		Pop valve	Center of area	17
88	6	100	1.370	1.362	−0.6				
89	5	100	1.372	1.377	+0.4				
	17					−0.2			
		Average of 5 groups (14 runs and 87 shots) = + 0.42 per cent.							
		Screen at Station A. 8-in. flat electrode at Station C.							
7a	6	25	0.401	0.406	+1.3		Perforated pipe and pop valve	Center of gravity (eye)	18
16	6	100	1.395	1.416	+1.5				
17	5	40	0.620	0.637	+2.7				
	17					+1.8			
		Two 8-in. flat electrodes at Station A and Station C.							
122	5	50	0.716	0.746	+4.2		Pop valve	Center of area	19
128	5	100	1.307	1.345	+2.9				
132	5	25	0.439	0.455	+3.6				
	15					+3.6			

Table 13 shows the lengths and volumes used. Fig. 43 is a sample data sheet of the 1923 laboratory investigations.

154 A discussion of the various methods of computing the discharge from the curves will be found in Section III of this paper. In these tests on the 8-in. pipe the curves obtained were usually symmetrical, so that the axis of the theoretically correct center of gravity coincided with the easily determined center of area. Since the main object was to compare the results by weight with the results computed from this center of area as determined by the integrator dots, that method of computing curves was usually

TABLE 18

Length of steel pipe used, 22.53 ft. Size, 8 in. Volume, 8.130 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
			8-in. improved electrode at Station A. Screen at Station B.						
80	6	100	1.394	1.399	+0.4		Pop valve	Center of area	20
81	6	50	0.721	0.723	+0.3				
82	6	25	0.434	0.436	+0.5				
	18					+0.4			
100	10	100	1.401	1.392	−0.6		Pop valve	Center of area	21
101	5	50	0.719	0.715	−0.6				
102	6	25	0.438	0.440	+0.5				
104	7	100	1.404	1.411	+0.5				
	28					−0.1			
109	5	25	0.424	0.428	+0.9		Pop valve	Center of area	22
110	5	50	0.722	0.727	+0.7				
113	5	100	1.400	1.395	−0.4				
	15					+0.4			
108	5	25	0.424	0.425	+0.2		Perforated pipe	Center of area without drag	23
111	4	50	0.722	0.712	−1.4				
112	5	100	1.400	1.380	−1.4				
	14					−0.8			
108	5	25	0.425	0.418	−1.4		Perforated pipe	Center of area with drag	24
111	4	50	0.722	0.692	−4.1				
112	5	100	1.400	1.328	−3.9				
	14					−3.1			

Average of 4 groups, consisting of 13 runs and 75 shots = − 0.01 per cent.
Omit group 24 on account of method of computation.

TABLE 19

Length of steel pipe used, 22.53 ft. Size, 8 in. Volume, 8.130 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
			8-in. flat electrode at Station A. Screen at Station B.						
129	10	100	1.304	1.299	−0.4		Pop valve	Center of area	25
130	5	50	0.708	0.707	−0.1				
131	5	25	0.439	0.441	+0.5				
134	5	25	0.439	0.440	+0.2				
	25					−0.04			
			Pipe 23.90 ft. long. Volume = 8.626 cu. ft. Signal on pump for introduction. Screen at Station B.						
91	5	100	1.373	1.374	+0.1		Pop valve	Center of area	31
93	5	50	0.719	0.713	−0.8				
96	5	25	0.427	0.428	+0.2				
	15					−0.17			

Average of 2 groups (7 runs and 40 shots) = − 0.09 per cent.

			Two 8-in. flat electrodes at Station A and Station C.						
125	5	50	0.716	0.717	+0.1	+0.1	Perforated pipe	Center of area with drag	38

Two errors balance each other.

TABLE 20

Length of steel pipe used, 20.91 ft. Size, 8 in. Volume, 7.548 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Two 8-in. improved electrodes at Station A and Station C.									
74	5	25	0.447	0.445	−0.4		Pop valve	Center of area	26
78	6	100	1.381	1.386	+0.4				
117	5	100	1.398	1.388	−0.7				
121	5	50	0.721	0.719	−0.3				
	21					−0.2			
97	5	25	0.456	0.454	−0.4		Pop valve	Center of area	27
98	6	50	0.732	0.726	−0.8				
99	10	100	1.400	1.400	0.0				
103	5	100	1.404	1.408	+0.3				
	26					−0.2			
75	5	25	0.447	0.448	+0.2		Perforated pipe below 8-in. ell.	Center of area without drag	28
79	5	100	1.381	1.368	−0.9				
	10					−0.4			
105	5	100	1.405	1.404	−0.1		Perforated pipe above 8-in. ell.	Center of area without drag	29
106	5	50	0.725	0.724	−0.1				
107	6	25	0.423	0.425	+0.5				
	16					+0.1			
105	5	100	1.405	1.363	−3.0		Perforated pipe above 8-in. ell.	Center of area with drag	30
106	5	50	0.725	0.718	−1.0				
107	6	25	0.423	0.418	−1.2				
	16					−1.7			

Average of 4 groups (13 runs and 73 shots) = − 0.16 per cent.
Group 30 omitted on account of method of computation.

TABLE 21

Length of steel pipe used, 20.95 ft. Size, 8 in. Volume, 7.560 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Traverses									
Screen at Station A. ¾-in. electrode at Station C.									
13	41	25	0.416	0.420	+1.0		Pop valve	By equal-area method	32 Hor.
14	42	50	0.730	0.726	−0.6				
15	41	100	1.394	1.378	−1.1				
3	124					−0.2			
47	35	100	1.399	1.402	+0.2		Pop valve	By equal-area method	33 Vert.
48	33	100	1.398	1.394	−0.3				
50	31	25	0.409	0.410	+0.2				
3	99					+0.1			

Average of 2 groups, 6 traverses = − 0.10 per cent.

Screen at Station A. ¾-in. electrode at Station C, held at a fixed point which was computed from the above traverses.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
55	10	100	1.400	1.410	+0.4		Pop valve	Center of area	34 Vert.
57	11	50	0.727	0.727	0.0				
58	10	25	0.439	0.443	+0.9				
3	31					+0.4			
59	10	100	1.397	1.423	+1.9		Pop valve	Center of area	35 Hor.
60	10	50	0.723	0.729	+0.8				
61	10	25	0.434	0.426	−1.8				
3	30					+0.3			

Average of 2 groups, 6 points = + 0.37 per cent.

followed. However, other methods of computation, such as curves by other meters, using peaks of curves, center of gravity of curves as determined by eye, with and without the drag at the ends of the curves, as well as the curves and dots made with poor salt-injection pipes, such as an open-ended pipe and a perforated pipe instead of a pop valve, were all used to determine the amount of variation.

155 As in the laboratory investigations in 1922, when the salt velocity method was compared with results by weir and

TABLE 22

Length of steel pipe used, 9.77 ft. Size, 8 in. Volume, 3.526 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
		Two 8-in. improved electrodes at Station A and Station D.							
114	5	100	1.406	1.410	+0.3	+0.3	Pop valve	Center of area	36
		Two 8-in. flat electrodes at Station A and Station D.							
122	5	50	0.716	0.744	+4.0		Pop valve	Center of area	37
123	5	50	0.716	0.742	+3.3				
127	5	100	1.307	1.331	+1.8				
128	5	100	1.307	1.367	+4.6				
132	5	25	0.439	0.454	+3.4				
133	5	25	0.439	0.454	+3.4				
	30					+3.4			
125	5	50	0.716	0.717	+0.1	+0.1	Perforated pipe	Center of area with drag	38
		Screen at Station A. 8-in. flat electrode at Station D.							
134	5	25	0.439	0.454	+3.7		Pop valve	Center of area	39
135	5	100	1.300	1.350	+3.8				
138	5	50	0.709	0.739	+4.2				
139	5	25	0.429	0.448	+4.4				
	20					+4.0			

On run No. 125, two errors, flat final electrode and computing curves with drag (one plus and one minus), cancel each other. Not reliable.

venturi meter, and the method of computing curves by fixing the axis of the center of gravity by eye gave results varying less than one per cent from the true Q, that method of computation applied to these 1923 tests gave results within one-half of one per cent from the results by weight.

156 Traverses were made across the pipe in both vertical and horizontal positions. The discharge was computed by plotting velocities at different positions in the pipe and using the equal-area method for computing the total discharge. It will be noted that these velocities are not instantaneous velocities at the electrode, but are average velocities between the introduction and the final electrodes. Fig. 44 shows sample traverse curves.

Results

157 Summaries of the results of these tests are given in Tables 14 to 23, inclusive. Analyses of these summaries are given in Table 25. All averages are weighted for number of shots per run.

158 Summarizing all of the tests shows that 36 groups, consisting of 123 runs and 1012 shots, indicated discharges which differed from the discharge by weight by only 0.05 per cent. This

TABLE 23

Length of steel pipe used, 11.14 ft. Size, 8 in. Volume, 4.022 cu. ft.

Run No.	No. of Tests	Gate, per cent	Q, Cu. Ft. Per Sec. by weight	Q, Cu. Ft. Per Sec. by salt	Salt, per cent	Average, per cent	Salt Injection	Computation	Group
Two 8-in. improved electrodes at Station D and Station C.									
115	5	100	1.406	1.416	+0.7				
116	5	100	1.407	1.404	−0.2		Pop valve	Center of area	40
120	5	50	0.721	0.715	−0.8				
	15					−0 1			
8-in. improved electrode at Station D. Screens at Station B. Pipe, 12.76 ft. long. Volume, 4.605 cu. ft.									
118	5	100	1.398	1.387	−0.8		Pop valve	Center of area	41
119	5	50	0.721	0.722	+0.1				
	10					−0.4			
Two screen electrodes at Station A and Station E. Steel pipe (8 in., 6 in., and 4 in. diam.) 30.20 ft. long. Volume, 9.913 cu. ft.									
135	5	100	1.300	1.305	+0.4				
138	5	50	0.709	0.711	+0.3		Pop valve	Center of area	42
139	5	25	0.429	0.432	+0.7				
	15					+0.5			
8-in. flat electrode at Station A. Screens at Station E. Pipe, 30.16 ft. long. Volume, 9.901 cu. ft.									
124	5	50	0.716	0.713	−0.4		Perforated pipe	Center of area with drag	
123	5	50	0.716	0.709	−1.0				43
127	5	100	1.307	1.299	−0.6		Pop valve	Center of area	
133	5	25	0.439	0.442	+0.7				
	20					−0.3			

Average of 4 groups (12 runs and 60 shots) = − 0.09 per cent.

is practically an exact check. This summary omits trial shots and all runs with known sources of error, such as flat parallel plates for the final electrodes, computations from the peaks of curves or with the drag on the curves using poor designs of salt-injection pipes.

159 The errors when using one pair of so-called "flat" final electrodes in a circular pipe varied from +2 to +4 per cent at different stations, due to giving the fast water in the center of the pipe a greater proportional length of electrode than the slow water near the walls of the pipe. The traverse curves in Fig. 41 show that the velocity of the center water averages 13 per cent greater

than the velocity of the water near the wall of the pipe. Other traverses on larger pipes of uniform diameter indicate a difference of from 7 to 12 per cent. An improved electrode was designed to correct this error. (See Fig. 40.)

160 The errors when using a perforated pipe for the salt injection and computing curves with all the drag included, vary from − 1 to − 3 per cent. The term "drag" is used to denote the dragging out of the pen curves as they approach the final appearance of the salt. With open or perforated distribution pipes this drag is caused by the continuing flow or oozing of the salt solution after the controlling valve has been closed. For a comparison of curves using a perforated-pipe introduction and showing a pronounced drag and of curves using a pop valve for the introduction and showing little or no drag, see Fig. 41, tests 988 and 989. Run 125, using a perforated pipe for the salt injection and one pair of flat electrodes at either stations A, C, or D, gave results only 0.1 per cent in error, but this was because the errors due to those two sources balanced.

TABLE 24 SUMMARY OF ALL TESTS SHOWN IN DETAIL IN TABLES 14 TO 24

For 36 groups, 123 runs, and 1012 shots, Q by salt differs from Q by weight by + 0.05 per cent. This omits trial runs, runs with flat final electrodes, and runs which have been computed with known errors for purposes of comparison, such as peaks and drags.

Analysis for gate opening shows full gate slightly more accurate, but the difference is too small to consider.

Analysis for station at which final electrode is located shows nothing conclusive.

Analysis for Type of Final Electrode

Electrode	Groups	Runs	Shots	Salt per cent
Screen	21	80	517	− 0.01
Flat	4	16	82	+ 3.23
Improved	11	31	180	+ 0.06
Traverse	4	12	285	− 0.09
	40	139	1094	

161 Analyzing the summaries for the various gate openings shows that the results are slightly better for full gate, but the difference is too slight to be of importance or conclusive.

162 Analyzing the summaries for the various stations used shows nothing conclusive except the accuracy of the method for any section of pipe. The results between A and E (+ 0.03 per cent), a section including 8-in., 6-in., and 4-in. pipe, were fully as accurate as when a section of 8-in. pipe alone was used.

163 Analyzing for the various electrodes shows that two screen electrodes gave the closest results with the results by the short traverse electrodes, and the results by the long improved electrodes almost equally close, while the flat final electrodes introduced a three per cent error.

164 Another method of correcting the error caused by using one pair of parallel plates for final electrodes in a circular pipe was also used. Two pairs of parallel plates were used, the upper one

placed a considerable distance below the salt-introduction station. The section of penstocks between electrodes and the center of gravity of the two salt curves were used in computations. This method has been used successfully on commercial tests.

Conclusions

165 These tests proved that when properly conducted the salt velocity method of water measurement checks the discharge by weight, which is the most accurate known method of measuring water. For short pipes the following items of apparatus and methods of computation were proven or confirmed:

a That a tight quick-closing pop valve is most suitable for salt injection.

b That other methods of salt injection can be used, but correct results are obtained only by applying an arbitrary and consequently inaccurate correction.

c That it makes very little difference what form of electrode is used at the introduction when the salt-injection pipe is very close.

d That screens or grids are ideal for both introduction and final electrodes.

e That an improved plate construction with proper spacing for the final electrodes gives very accurate results.

f That a traverse with short final electrodes gives equally accurate results.

g That the same electrodes held at a fixed point, determined by the traverse, also show accurate results.

h That computations based on the center of area as determined by a watt-hour meter and a series of jump-spark dots simplify the work and give accurate results.

i That tests can be repeated indefinitely and still check.

j That the apparatus and methods of computation used in the commercial tests were proven to be theoretically and practically correct, and the results obtained were accurate and reliable.

166 And finally, the authors believe that the salt velocity method of water measurement is correct in theory and in practice, that it is applicable to any form or size of flume, pipe, or penstock, and that, in a few years, its simplicity and accuracy will make it an accepted standard method of water measurement.

167 The Laurentide Power Company of Grand Mere, Quebec, made it possible to carry on a large part of these investigations. The authors wish to thank Mr. John S. Riddile for his interest and assistance, and also Mr. Arthur K. Ingraham for his work with us in the earlier stages of development.

No. 1903

THE GIBSON METHOD AND APPARATUS FOR MEASURING THE FLOW OF WATER IN CLOSED CONDUITS

AS APPLIED IN TESTING THE EFFICIENCIES OF WATER WHEELS IN HYDROELECTRIC POWER PLANTS

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Non-Member

The purpose of this paper is to explain a new method of determining the rate of discharge or quantity of water flowing in a pipe or other closed conduit and to describe the apparatus used for the practical application of this method in testing the efficiencies of water wheels in hydroelectric power plants.

The theory of this method of measurement is based on Newton's second law governing the relation between the mass and motion of bodies, and on the principles attributed to Joukovsky relating the rise of pressure in a column of water to the retardation of its velocity.

The paper explains the procedure in the field and describes the apparatus employed and the manner of recording, delineating, and measuring the pressure-time diagram from which the discharge is calculated. The fundamental principles and theory are briefly discussed and a complete typical numerical example is computed. In conclusion, the results of tests made at Cornell University to determine the accuracy of the method are given.

IN A FORMER PAPER[2] by the author it was shown how the excess pressure in penstocks caused by the gradual closing of turbine gates might be determined from Joukovsky's theory of water hammer. The present writing is intended to show how this excess pressure may be used to determine the rate of discharge or quantity of water flowing in the pipe at the moment the gates

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[2] Pressures in Penstocks Caused by the Gradual Closing of Turbine Gates. Trans. A.S.C.E., 1920.

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began to close. Experiments will be described which have been made to determine the accuracy of such measurements when applied to the testing of water wheels in hydroelectric power plants.

2 This is a new method of water measurement and it has required the invention of new apparatus for its practical application. If any apology is considered necessary for introducing it at this time when so many well-tried methods and devices are already in general use, it may be found in the fact that some important advantages have already been obtained through its application to the testing of water wheels, where the inconvenience and expense of any of the older methods would have been objectionable. These advantages have made it possible to make tests at frequent intervals, at slight expense, to determine the operating condition of water wheels without seriously interfering with their commercial operation. It may also be added that in point of accuracy, measurements by the new method have been found remarkably precise when tested in accordance with the dictum of Clemens Herschel, the distinguished hydraulic engineer that "The only standard water-gaging apparatus is a tank or reservoir. Weirs, orifices, venturi tubes and meters, and other water-gaging apparatus and methods can become truly rated only by comparison with tank measurements and thus thereafter competent to give reliable service."[1]

3 The new method of water measurement herein described has been in practical use for upward of three years. As a result of this experience, it may confidently be asserted that the method combines the qualities of accuracy, convenience, and economy, although it is not to be expected that the full extent of its usefulness can be determined until after a considerable period of time.

4 To treat the development of the method chronologically or to describe the various steps that were taken from the moment of its conception to the present time would surely be tedious and uninteresting. Moreover no particular value would now be attached to such a record. It will be much more satisfactory first to explain the successive operations to be performed during the efficiency test of a water wheel by this method, then to describe the apparatus used, and finally to examine the principles and theory on which the method is based and the experiments which have been carried out to determine its accuracy in practical cases. In Appendix No. 1 will be found a complete typical example and a discussion of some of the various factors and special problems which may sometimes require consideration.

[1] An Improved Form of Weir for Gaging in Open Channels. Trans. A.S.M.E. 1920.

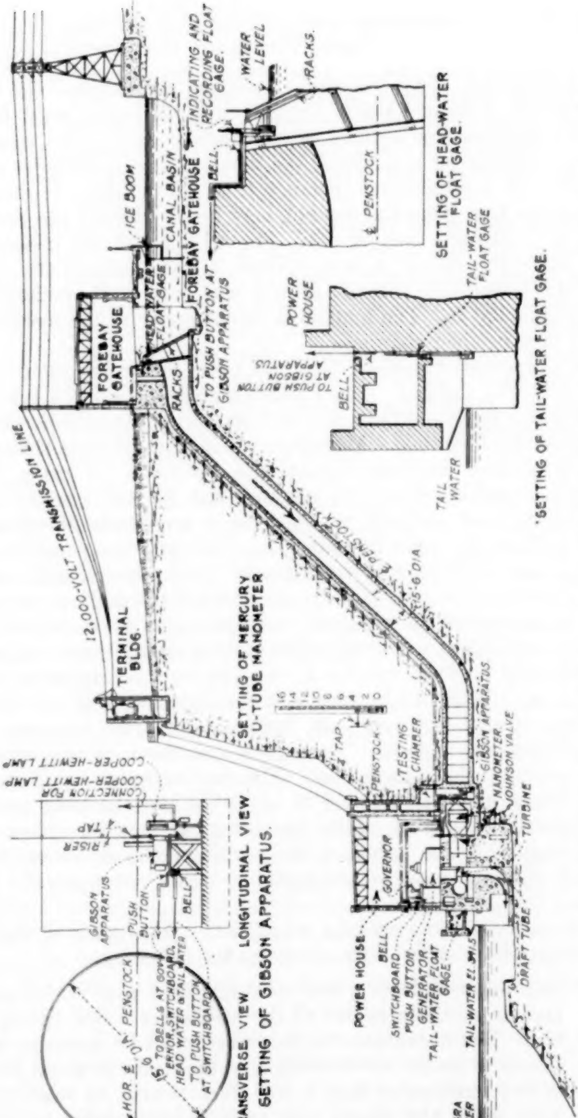

FIG. 1 CROSS-SECTION OF TYPICAL HYDROELECTRIC POWER PLANT, SHOWING LOCATION AND SETTING OF APPARATUS AND EQUIPMENT FOR EFFICIENCY TESTS

Field Work

5 *Preliminary.* Two essential conditions are required for the measurement of water by this method, first, the water must flow through a pressure pipe or other closed conduit; and second, means must be available for controlling the flow, such as a valve or turbine gates, at a point some distance from the intake. It makes no difference how large the conduit may be or whether the cross-section is of uniform area or not. The accuracy of the measurements, however, depends to some extent on the length of the conduit, and it is desirable that this should be at least 50 ft., preferably not less than 100 ft. The method does not apply to the measurement of flow in open channels.

6 It is evident, therefore, that not every hydraulic plant can be tested by this method. Fig. 1 shows a cross-section of a typical plant, the physical conditions of which are suitable for the simplest application of the method. Measurements of water flowing through more complicated waterways may also be made by using the differential application of this method, but for the present the discussion will be confined to the simple type here illustrated. In such a plant the water flows out of a forebay into a penstock at the lower end of which is a single modern hydroelectric unit comprising a turbine and generator equipped with a hydraulic governor. At the upper end of the penstock is a sluice gate or valve and at the lower end just at entrance to the turbine casing there is sometimes installed another valve. To set the unit in operation the turbine gates are opened by means of the governor and the water then flows through the penstock, turbine, and draft tube and is discharged into the tailrace. The unit is shut down by closing the turbine gates by the same means. The flow of water in the penstock is therefore under control at all times and the time taken in the opening or closing of the turbine gates may be varied at will by regulating the speed of the stroke of the governor, the operation of which may be accomplished either automatically or by hand.

7 The apparatus required for an efficiency test, parts of which will be described in detail later, comprise the following:

a A headwater-indicating and recording float gage which will give an accurate record of the headwater levels during a test. The instrument for this work must be specially designed to make one revolution of the drum in about four or five minutes so that it will show clearly, to relatively large scale, any surges that occur in the forebay during or immediately following the closing of the turbine gates.[1]

b A tailwater float or staff gage for observing tailwater levels. It is not essential that this should be a recording instru-

[1] When differential diagrams are used, a simple staff gage is sufficient for determining headwater levels.

ment as it is usually quite satisfactory to make periodic readings of the gage.

c The apparatus for obtaining pressure-time diagrams, which has been called the Gibson Apparatus,[1] and which is attached to the penstock at any convenient point by means of a 1/4-in. connecting pipe as illustrated in Fig. 1.

d A pressure gage or piezometer for measuring the head acting on the turbine.

e Electrical equipment for measuring the output of the generator, including standardized calibrated wattmeters, current and potential transformers, and ammeters and voltmeters for determining the power used in excitation. It is assumed that the efficiency of the generator will have been previously determined.

f A signal system (usually electric bells) for communicating with the observers at the forebay, tailrace, pressure gage, Gibson Apparatus, and electrical instruments and with the operator at the governor.

8 The items enumerated above include, of course, not only the apparatus necessary to measure the water, but also the equipment for measuring the power output of the turbine and for determining its efficiency. The location and setting of the apparatus and equipment is shown in Fig. 1.

9 *Procedure.* The various operations to be performed in the field to obtain the data from which the final computations are made are as follows:

10 It is first necessary to obtain the physical dimensions of the pipe line from its intake at the upper end to the point of attachment of the Gibson Apparatus. This may be done from the drawings, but where necessary the dimensions should be checked by measurement in the field. When the penstock is made up of pipes of various sizes, the length and area of each size should be determined and where a variable section exists, as for instance, at a flaring mouthpiece, the lengths and areas of successive sections a short distance apart should be taken so that the whole may be properly integrated. The total length along the center line of the penstock from the apparatus to the face of the mouthpiece should be accurately determined and should check with reasonable accuracy the sum of the lengths of the various parts.

11 The personnel engaged at the various posts may be selected from the engineering and operating staffs engaged at the plant and should be directed by one man who has no specific duties to perform in order that he may supervise all the operations from time to time and thus be able to review the work as a whole. In addition one observer will be stationed at the headwater gage, one at the tailwater gage, and one at the pressure gage, or two if a

[1] Method and apparatus patented, Canada 1919 and 1920, United States Oct. 13 and Nov. 22, 1921. Patents applied for in foreign countries.

piezometer be used at entrance to the turbine case. One operator and sometimes an assistant will be required for the Gibson Apparatus. At least two observers will be necessary for the electrical readings and one man, usually a station operator, is required to operate the governor for controlling the turbine gates. An observer to record gate opening and read a vacuum gage on the draft tube of the turbine will be useful in important tests, but these observations are not always essential.

12 The signal system is controlled by the operator of the Gibson Apparatus and a prearranged system of signals enables him to synchronize the work of the various observers and to have the necessary operations performed at the required time.

13 After checking the elevations of the zeros of the headwater and tailwater gages, testing the pressure gage for measuring the head on the turbine, setting up the apparatus and signal system and inspecting the unit to be tested, etc., the operations to be performed are as follows:

14 The unit to be tested is usually synchronized with other units in the plant and allowed to run on steady load at the desired point with the gates fixed for several minutes. If desired, an artificial load of any kind such as a water rheostat may be used for testing, but in this case the unit will be out of commercial service, which may cause more inconvenience than allowing it to remain on the commercial load. In fixing the gates at the various percentages of gate opening where it is desired to determine the efficiency, it is important to have them perfectly steady so as not to set up disturbances in the penstock. For this purpose it is better to use the hand control, which is usually supplied in a modern governor, but if this is not available various devices may be employed to lock the automatic control valve of the governor so as to hold the gates steady for a few minutes at a time until it is necessary to release them.

15 When everything is in readiness a signal is given, say, "2 bells," and at that moment the various observers begin to read their instruments in the most effective manner. Thus observations are taken of the headwater and tailwater levels, pressure at entrance to the turbine case, gate opening, kilowatt output of the generator, voltage and amperage of exciting current, and the usual entries of time and other items worthy of note.

16 After these observations have been made for a period of two minutes, or longer if necessary, the operator of the Gibson Apparatus sets this instrument in motion and then gives a signal, say, "1 bell," at which all the observers mark their records and the governor attendant operates the governor so that the turbine gates are gradually and gently closed. The duration of this closing action will depend upon the length of the penstock. If it is short, a quite rapid closure, say, in three or four seconds, may be made. If the penstock is very long the duration of closure may take perhaps more than a minute. In a penstock from 200 to 400 ft. long

a satisfactory closure may be made in from eight to sixteen seconds. It is not at all necessary that the closure should be made at a uniform rate, although, of course, it is better that an effort should be made to perform this operation as smoothly as possible.

17 As the turbine gates close, the power output of the unit of course diminishes and its load is picked up by the other units operating in synchronism with it. When the gates are fully closed the unit continues to revolve at normal speed, being then operated as a synchronous motor by the current in the circuits to which it is connected.

18 The object of closing the turbine gates in this manner is to produce a change of pressure in the penstock which will occur when the flow of water therein is brought to rest. The Gibson Apparatus (which will be described in detail later) is devised to obtain a precise record of this change of pressure in the form of a diagram of which Fig. 7 is a typical example. On this diagram *A* marks the beginning of the record. At *O* the pressure begins to rise as the turbine gates are being closed. At *K* (the determination of which will be explained later) the gates are fully closed and from *K* to *C* the column of water is being restored to equilibrium in a series of damped harmonic oscillations or waves. At *Z* there is an interval of time sufficiently long to allow these oscillations to subside, and then at *F* there is a short record which registers the pressure in the penstock under the then existing hydrostatic conditions. It will be observed that narrow vertical spaces *D* where the record has been obliterated occur on the diagram at regular intervals. These intervals are records of time usually one second apart, or, speaking more correctly for reasons which will be explained presently, each pair of intervals measures exactly two seconds.

19 Upon the completion of this record — called the "pressure-time diagram," or more simply "the diagram" — a signal is given to the observers which terminates the "run" as the operations of the test at each load are called.

20 The unit is then loaded up again to the next point desired and everything is placed in readiness for the next run. These operations are repeated as often and at as many points as are desired. Usually it will be found that from 10 to 12 diagrams ranging from half-gate to full load will be sufficient. Each run will require from 10 to 15 minutes, so that if no difficulties arise the field work for one unit may be completed in from two to three hours, exclusive of time required in the preparation for the test and for the leakage test which will now be described.

21 *Leakage Test.* In this method of water measurement only the flow actually shut off is measured. If the whole flow is shut off, no other operations than those described above are necessary. It usually happens, however, that the valve to be closed or the turbine gates themselves are not perfectly tight and after closure some water leaks through. The measurement of this leakage is

not included in the diagram and must of course be separately determined and added to the quantity calculated from the diagram to obtain the total discharge. In a new modern unit the leakage is relatively quite small and its amount can be quickly and precisely determined in a number of ways. If the total discharge of the turbine is 1000 cu. ft. per sec. and the leakage is 20 cu. ft. per sec., an error of 5 per cent in determining the leakage results in a final error in the measurement or discharge of only about 0.1 per cent. Usually, however, the leakage may be determined with much greater accuracy than this and often it can easily be measured precisely. When the leakage is relatively large, greater care in its determination will be required if precise results are desired.

22 The leakage test as usually carried out is as follows: First the turbine gates are tightly closed, full operating pressure of the governor fluid being maintained in the regulating cylinders, and then the head gate at the intake of the penstock is closed so that the water in the penstock will run out through the clearance spaces around the turbine gates through which the leakage takes place. As the water is leaking out, readings are taken of the pressure in the penstock as indicated by the pressure gage at entrance to the turbine case, and the time interval between each reading is observed and recorded from a stop watch. These observations are continued until the water in the penstock has leaked out to a low level. The records of pressure and time thus obtained measure the rate at which the water has leaked out, and as the area of the penstock at any elevation is known, the quantity of leakage under any head may be readily calculated.

23 If the leakage through the head gate is too great to be estimated with sufficient accuracy to make the necessary correction, it may be calculated in a manner to be described later (see Appendix No. 1), or other means may be found of measuring it as precisely as desired. For instance, if there is a tight valve in the penstock at its lower end, this may be closed and the rate at which the water fills up the penstock may be determined in a similar manner to that described above for determining the rate at which it leaks out through the clearance spaces around the turbine gates.

24 While the foregoing describes the usual way in which the leakage factor may be determined, it is not intended to lay down any definite rules by which it must be found. The essential point is that it should be determined with sufficient precision in a given case, and it may safely be left to the common sense and resourcefulness of the engineer in charge of the test to do this in a satisfactory way.

Apparatus

25 In describing the field work of the test, reference has been made in Par. 7 to the apparatus and equipment used, the principal items of which are the following:

Headwater gage for recording headwater levels
Tailwater gage for observing tailwater levels
Pressure gage or piezometer for measuring the head acting on the turbine
Electrical instruments for measuring the power output of the generator
Signal system for communicating with the observers, and
The Gibson Apparatus for obtaining the pressure-time diagrams.

26 Fig. 1 is a general drawing showing the setting and arrangement of the apparatus and equipment, and also some details of the minor parts.

27 The headwater recording gage which is shown in Fig. 2 has been designed specially for use in the simple application of this method of testing. It comprises a brass float 8 in. in diameter surrounded by a well consisting of a short piece of 9-in. brass tubing mounted on a heavy plank which may be fastened in place in any convenient manner. A rod from the float is connected to a crosshead which slides between guides and is connected through a reducing block and tackle to a brass pencil as illustrated in Fig. 2. The vertical travel of the pencil may be made to correspond in any desired ratio to the variation in level of the float, but in the instrument here illustrated the pencil moves one-half the vertical travel of the float. The supports for the rod and pencil are attached to a box, the front end of which has been left open. This box contains a drum about 6 in. in diameter and 12 in. high which is operated by means of a spring motor and gearing designed so that the drum will make one revolution in from 3 to 5 min. For each run a sheet of coated paper 18 in. long by 11 in. wide is securely wrapped around the drum and receives the impression from the brass pencil which rests lightly on the surface of the paper and thus records to half-scale the motion of the float in the headwater. A clock mechanism and electric battery and magnet located inside the box operate a pointer which makes a mark every five seconds on the coated paper near its lower edge. The record of headwater level is thus combined with a record of time.

28 It is unnecessary to describe in detail the tailwater gage or the pressure gage or piezometer. For the former an ordinary staff gage is suitable, but a float gage with pointer is sometimes more convenient. The pressure gage should of course be accurately calibrated. The mercury piezometer is preferable for important tests and precise work.

29 As this paper is not specially concerned with the measurement of electrical power, an elaborate description of the electrical instruments required for measuring the output of the generator during the test will not be given. It will be sufficient to say that great care should be used in measuring the electrical output of the generator for precise testing of hydroelectric units. The usual

instruments to employ are wattmeter (preferably in duplicate), ammeter, and voltmeter for checking the power indicated by the wattmeter, power-factor meter, calibrated current and potential transformers, and, where the unit is separately excited, an ammeter and voltmeter for measuring the power supplied for excitation will also be required.

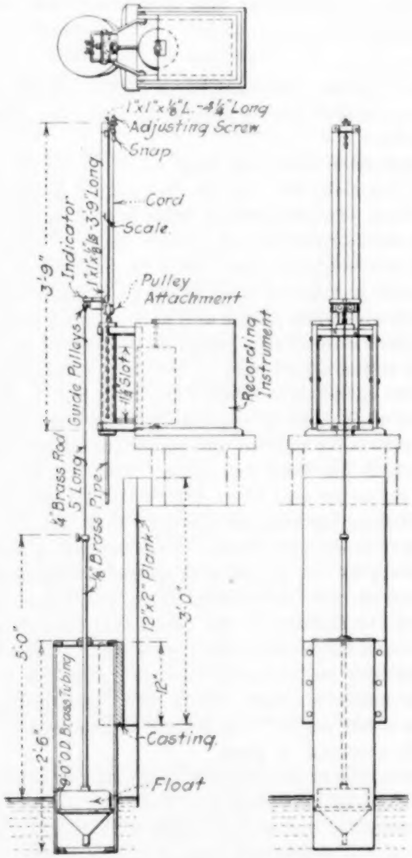

FIG. 2 HEADWATER RECORDING GAGE

30 The signal system may be left to the discretion of the engineer in charge of the test. Usually a system of electric bells will be sufficient, preferably a system operated with magneto because the bells may then be connected in series, in which case there is less danger of a run being spoiled by failure of one of the observers to receive his signal owing to trouble in one of the cir-

cuits. With the magneto set if one bell rings all must ring, or if one is out of order none will ring. A telephone system of signaling is quite satisfactory and very convenient when the distances between observers are great. When distances are short and when conditions will permit, a horn or whistle may sometimes be used.

31 The apparatus for obtaining the pressure-time diagrams will need to be described in detail. Photographs of the apparatus are shown in Figs. 3 and 4. It is also shown diagrammatically in Fig. 5, to which the letters in the following description refer.

32 *P* is a pipe or conduit in which the fluid, the rate of discharge of which is to be measured, is assumed to be flowing in the direction indicated by the arrow, and *V* is a valve or other means, such as turbine gates, for interrupting the flow of the said fluid. In the earlier forms of this apparatus an auxiliary device was used for recording the times of beginning and ending of the motion of the valve as shown in the detail view of the valve *V*. In the development of the new method of measurement, however, other means have been invented, as will be explained later, for determining the points of beginning and ending of the pressure-time diagram and the use of the auxiliary device referred to has been discontinued.

33 The apparatus is attached to the pipe *P* on the upstream side of the valve *V* at some point *B* by means of a short small pipe *C* in which a valve is placed near *B*. *D* is a glass tube of as nearly as possible uniform bore joined to the pipe *C* and to the riser pipe *E* — also as nearly as possible of uniform bore — by means of stuffing boxes and glands at *F* and *G*. Together the pipes *C* and *E* and the tube *D* form a U-tube. A quantity of mercury is poured into the top of the riser *E* or is otherwise introduced into the U-tube so that when the valve at *B* in the pipe *C* is opened the mercury column is depressed in tube *D* and rises in the riser pipe *E* in proportion to the pressure existing in the pipe *P*. Needle valves, as indicated, are provided at points 3, 4, and 5 for adjusting the height of the mercury in the tube *D* and for expelling air, etc. For higher heads an enlargement of the tube *C* is made to facilitate filling the U-tube with mercury. The difference in level between the tops of the mercury columns in *D* and *E* is then a measure of the pressure existing in the pipe *P*. Changes of pressure in *P* produce corresponding changes of level of the tops of the mercury columns in *D* and *E*.

34 Opposite the tube *D* is a box *H* containing a photographic lens and shutter *J* and a pendulum *K*. At the top of the pendulum is a setscrew and bar *L* for adjusting the length of the pendulum. The pendulum cord is thickened at its lower extremity, as shown, and the whole is enclosed in an upward extension of the box.

35 In the end of the box opposite *D* is a slot about as long as

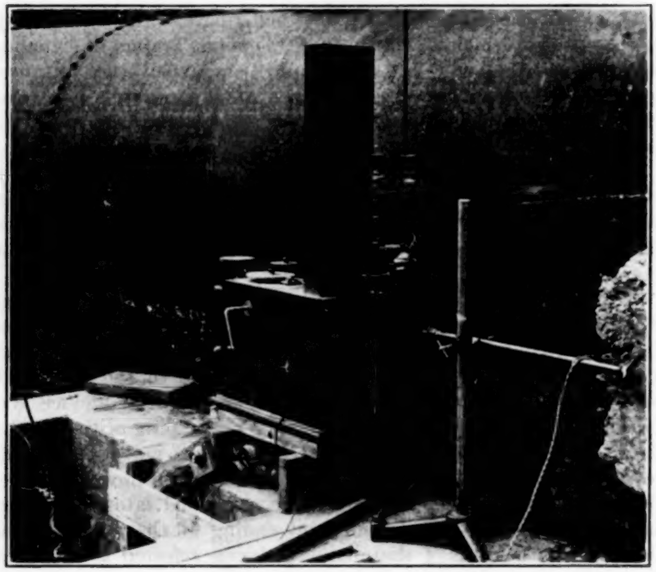

Fig. 3 Front View of Gibson Apparatus

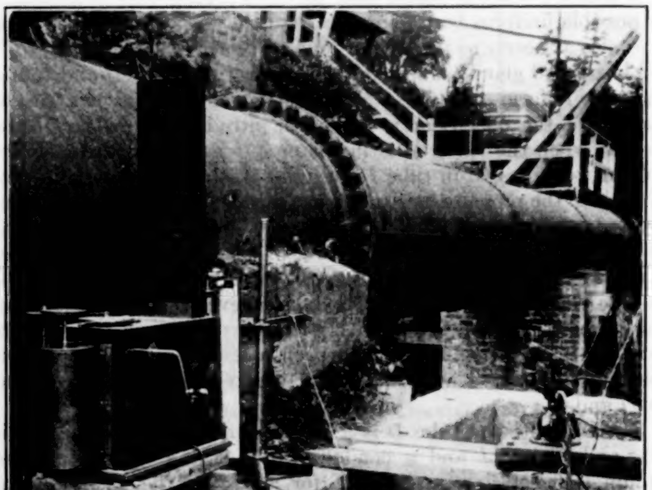

Fig. 4 Rear View of Gibson Apparatus

the tube D and narrower than the diameter of the tube. Between the box and D is a shrouding M which excludes the light, except that which enters through the tube D from the side directly opposite the slot referred to. This shrouding partly surrounds the tube D and has fastened to it, crosswise of the tube D, two fine cross-wires 1 and 2, a measured distance apart.

36 At the other end of the box is an opening in which a ground glass may be placed for observing a photographic image of the tube D cast on the glass by means of the lens J. The ground glass is for observation purposes only and may be removed and replaced by the revolving drum N set in a lightproof holder which

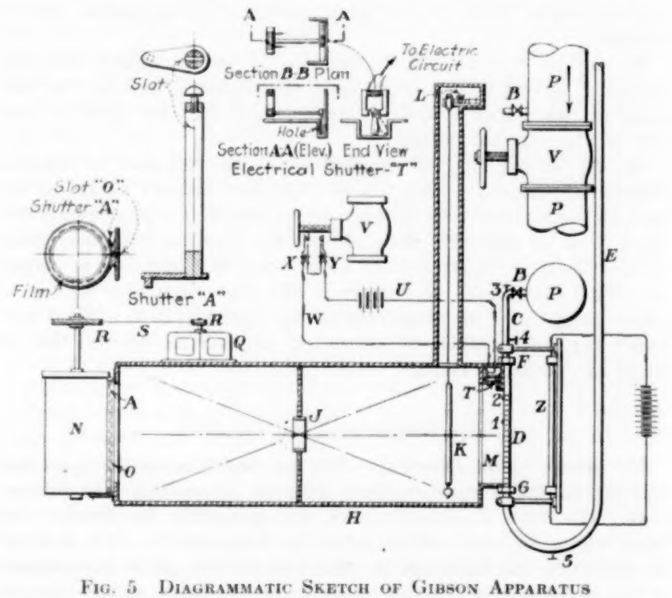

Fig. 5 Diagrammatic Sketch of Gibson Apparatus

may be attached to the end of the box H by means of guides and thumbscrews.

37 On the outside surface of the revolving drum N there may be attached, as required and in a dark room, a sensitized photographic film on which the pressure-time diagram is to be recorded. Photographic paper may be used, if desired, in place of the film, but usually will not give so good results on account of the shrinkage of the paper when developed.

38 On the side of the holder adjacent to the box there is a narrow slot, O, lengthwise of the holder, which may be opened or closed by means of the shutter A. This shutter is shown in detail in Fig. 5 and consists of a cylindrical bar through which a

slot has been cut. The bar fits snugly in a hole bored in the casting supporting the lightproof holder and may be revolved so that the slot in the bar will come opposite the slot O in the holder when it is desired to allow the light to pass through to the film on the drum N, or it may be turned 90 deg. from this position when it is desired to shut off the light from the film. The shutter may be held in either position by means of an arm attached to the bar and secured by a spring attached to the holder.

39 The drum N is revolved at uniform speed by means of a mechanical motor Q through pulleys R and belting S or through other means, its speed being regulated by a governor and flywheel.

40 In case the apparatus cannot be used in daylight, an electric mercury-vapor lamp Z or other means of illumination will be required.

41 As previously stated, the use of the auxiliary electrical shutter T shown in Fig. 5 has been discontinued, and for the purposes of this paper it will be unnecessary to refer again to this part of the original design.

42 In setting up the apparatus for the purpose of making differential diagrams, the top of the riser instead of being left open to the atmosphere is connected by means of a pipe, preferably about 1 in. in diameter, to a piezometer opening located at some suitable distance upstream from the point B. Mercury is poured into the U-tube until its surface in the glass tube D is at the required height and the remainder of the piping system is filled with water, care being taken to see that all air is excluded and that all joints are perfectly tight.

The Pressure-Time Diagram

43 *Recording the Diagram.* The manner of using the apparatus and of making a pressure-time diagram is explained as follows:

44 The final equation which will presently be derived and from which the rate of discharge is determined is $Q = KA/SF$, in which Q is the discharge in cubic feet per second, K the constant of the apparatus, S the time scale of the diagram, and F the pipe or conduit factor.

45 The pipe factor F of a simple pipe of uniform cross-section, such as the pipe P, Fig. 5, is the length from the point B to the upper end of the pipe divided by its area. When the pipe is made up of several lengths in series l_1, l_2, l_3, etc., of different areas a_1, a_2, a_3, etc., then the pipe factor F is

$$\sum \frac{l}{a} = \frac{l_1}{a_1} + \frac{l_2}{a_2} + \frac{l_3}{a_3} \text{ etc.}$$

The pipe factor is determined first.

46 Next the ratio R of the areas of tubes D and E is obtained. This may be done in several ways, only one of which need be mentioned. For this purpose a calibrating rod is provided

with the equipment. This rod is made up of several lengths of copper wire surrounded by, but insulated from, a small brass tube or sheath. These lengths may be screwed together and are so designed that connection is made between each length of copper wire without making contact with the brass sheath.

47 For the purpose of calibrating for R the Gibson Apparatus is set up either in the laboratory or in the field at the place where the tests are to be made and the tubes are filled with mercury as usual. By means of a small pump, or by admitting pressure in successive increments from the penstock, the mercury is depressed in D which causes it to rise in E. If the work is done in a laboratory, intermediate sections of the riser are removed so that the part of the riser E in which the mercury changes position when depressed in D is the same part as that in which the change will take place when the apparatus is attached to the penstock. It is obvious of course that when the tube D is attached to a pipe or penstock the static head existing in the pipe will force the mercury to a higher level in E and additional lengths of riser are necessary. The changes of level corresponding to changes of pressure during a test then take place in the glass tube and the sections of riser which have been calibrated. In practice it will be convenient to attach the apparatus at some point on the pipe line where the head will not be so great as to require an excessively high riser.

48 The calibration is then made by wrapping a film on the drum in the usual way and exposing a short section of it at a time, as hereinafter described, so as to obtain a record of the elevation of the mercury in tube D at different stages, and at each stage the calibrating rod is inserted in the top end of the riser E until contact is made between the wire in the rod and the mercury. A bell and some batteries are connected in circuit with the calibrating rod and the riser so that when contact is made between the rod and the mercury the bell rings. By this means it is possible to determine when the rod is first touching the surface of the mercury. The distance from the top of the riser to the surface of the mercury is then read on the vernier attached to the calibrating rod and recorded. After a number of exposures have been made with the mercury depressed to various points in the tube D and the corresponding position of the mercury in E is determined, the film is removed from the drum and developed and a print made as illustrated in Fig. 6. The record thus obtained when related to the height of mercury in the riser E gives the ratio of the change in E for a given change in D. This method of calibration also includes any correction that might be necessary for parallax or variation in scale.

49 When increases of pressure occur in pipe P it is evident that the mercury in D will fall $1/R$ times the distance it rises in E, and the total change of pressure in feet of water will be $\frac{1}{12}[M(R+1)-1]$ times the distance in inches the mercury falls in D, where M is the specific gravity of mercury. Similarly, when

decreases of pressure occur in P the mercury will rise $1/R$ times the distance it falls in E, and the total change of pressure in feet of water will be $\frac{1}{12}[M(R+1)-1]$ times the distance in inches the mercury rises in D. When the apparatus is connected to the pipe P for the purpose of making differential diagrams as described in Par. 42, the total change of pressure in the pipe P in feet of water will be $\frac{1}{12}(M-1)(R+1)$ times the distance in inches that the surface of the mercury changes in the tube D. The sizes of the tubes D and E for any particular case are selected so that the total motion of the mercury in D is limited to the length of the tube D. A record of the position of the top of the mercury column in the tube D during any time is therefore a record of the pressure existing in the pipe P during that time. When the flow of water in the pipe P is interrupted, a change of pressure occurs. The change of pressure affects the level of the mercury column in tube D as already mentioned, and a record of this change of level is obtained as follows:

50 The lens J having been adjusted and permanently fixed so that a clear image of the top of the mercury column is projected on the ground-glass screen at the end of the box, the latter is permanently removed so that the film holder may be inserted in the guides. A sensitized photographic film is wrapped around the drum N and fastened in place, the belting is slipped over the pulleys, and the motor wound up ready to start.

51 When all is ready the shutter A attached to the film holder is opened, the motor Q is started, and, just prior to the action of interrupting the flow in pipe P, the shutter of lens J is opened and the pendulum K is set in motion by pressing the pneumatic bulb attached to these parts. At this moment the flow in pipe P is gradually interrupted by means of the valve V or by any other means. As the flow is being interrupted, the rise of pressure produced thereby causes the mercury column in the tube D to move downward. The amount of its depression is a measure of the change of pressure in the pipe P, and the motion of the top of the mercury column is photographed on the revolving film by the means above described, thus furnishing a record of the change of pressure that took place in the pipe P as the flow was shut off.

52 During this operation the pendulum K keeps swinging to and fro, and on account of the enlargement of the cord already mentioned, the light passing through the tube above the mercury is cut off at each swing of the pendulum. This causes a break in the record on the film at regular intervals corresponding to the time period of the swing of the pendulum, which is usually adjusted to exactly one second. If the pendulum is not centered exactly with the slot, the intervals between breaks in the record will be alternately long and short. For this reason, as explained in Par. 18, it is better to use each pair of intervals which will exactly measure two seconds.

53 After the flow has been completely shut off, the oscillations

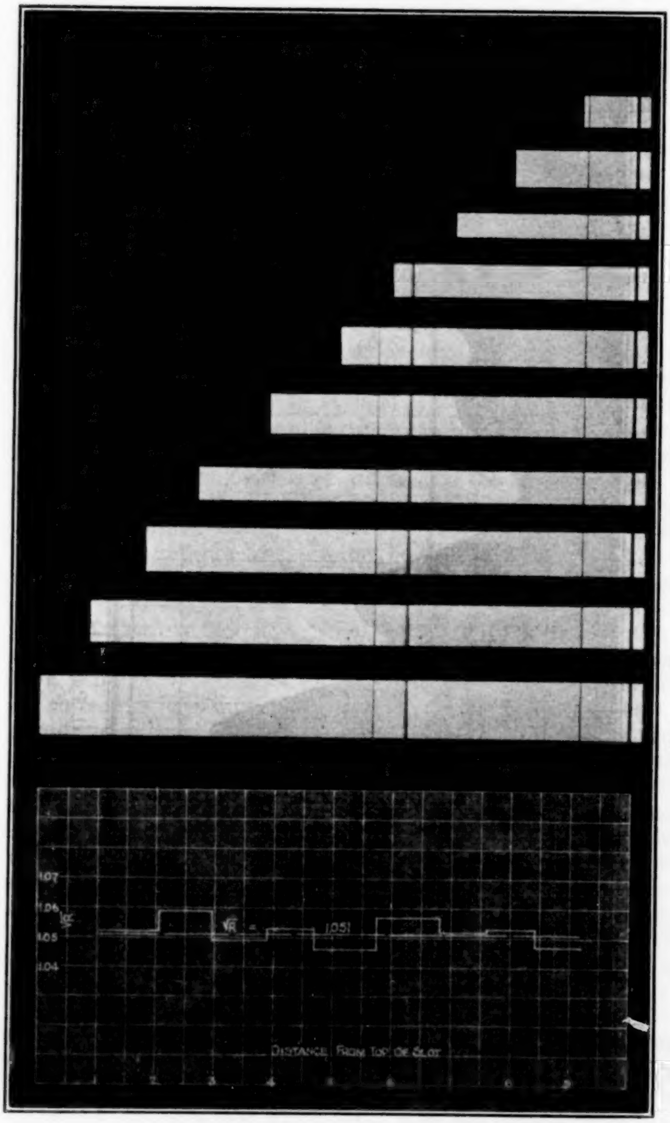

FIG. 6 TYPICAL CALIBRATION DIAGRAM

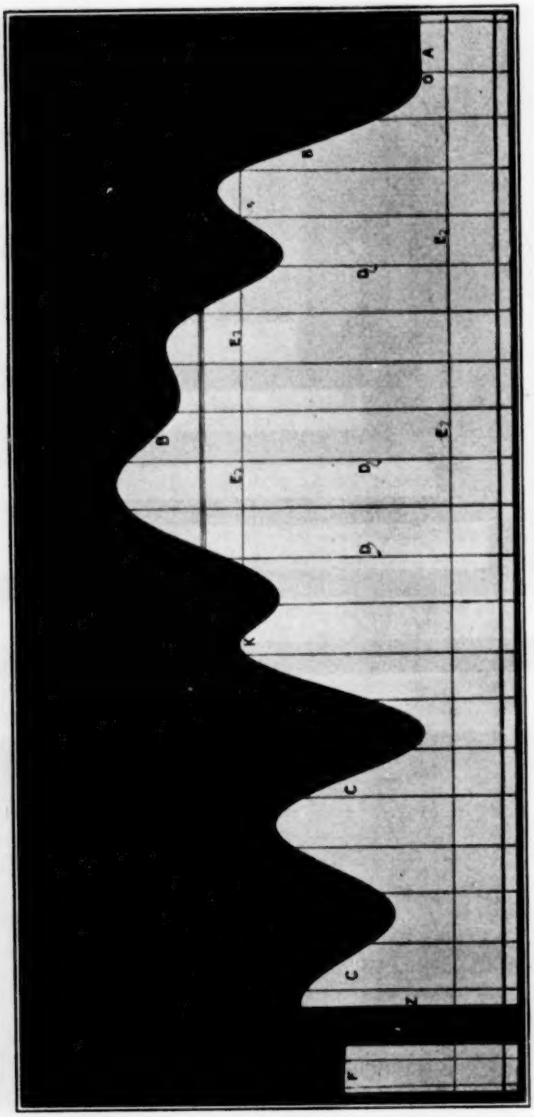

FIG. 7 TYPICAL PRESSURE-TIME DIAGRAM BEFORE DELINEATION

of the mercury column are recorded for a few seconds. After the oscillations have ceased and quiescent conditions exist in the pipe *P*, a short exposure is made to obtain a record of the static pressure in that pipe. Then the shutters are closed, the film removed and developed, and prints made. The result is shown on the typical pressure-time diagram, Fig. 7, on which the line *ABBKCC* represents the varying level of the top of the mercury column in tube *D*. The *D*-lines are the breaks in the record at regular intervals caused by the swinging pendulum, and the lines *E*, *E* are the photographs of the cross-wires 1 and 2 shown in Fig. 5, which are a measured distance apart, and from which the scale (usually exactly full size) of the diagram is shown. The edge *F* is a record of the level of the top of the mercury column corresponding to the static pressure in the pipe *P*.

54 *Delineating and Measuring the Diagram.* The diagram, Fig. 7, having been obtained in the manner described in the foregoing section, is treated as shown in Fig. 8 as follows. It will be assumed for simplicity in explaining the calculations that the gates were perfectly tight when closed so that the flow was completely shut off when the diagram was made. The numerical example which will be given later will include the leakage factor.

55 1 — Horizontal lines are drawn across the diagram, coinciding with lines *F* and *A*, respectively. If *A* is wavy due to slight variations in pressure of the water flowing in the penstock, the mean position between crest and valley of the wave may be used.

56 2 — As the diagram is produced by means of an exposure through a narrow slot in the film holder, the width of slot will cause the diagram to be slightly larger than it would be if drawn by a point or by exposure through a slot of infinitesimal width. To correct for this lines may be drawn as shown at *BB* and *CC* at a distance equal to the width of slot from the edges of the pressure line so that when thus drawn these lines, together with the other edges of the diagram, delineate the pressure line that would be drawn if the slot were of infinitesimal width. A little study will make it clear that the width of slot is drawn only on either the right-hand or left-hand edges of the diagrams but not on both, depending on the direction of rotation of the drum. If desired, this correction for width of slot may be made by simply multiplying the width of the slot by the sum of the vertical heights of the corresponding edges of the pressure line. Usually this will be more satisfactory.

57 3 — The point *K* which marks the end of the diagram, or more specifically the end of that part of the diagram which represents the impulse produced during the closure of the gates, is determined from the record *CC* of the oscillations of the mercury column in the apparatus which occur after the turbine gates have ceased to move. These oscillations or waves are of uniform periodicity like the oscillations of a pendulum, and the horizontal distances measured in time intervals between the vertical lines pass-

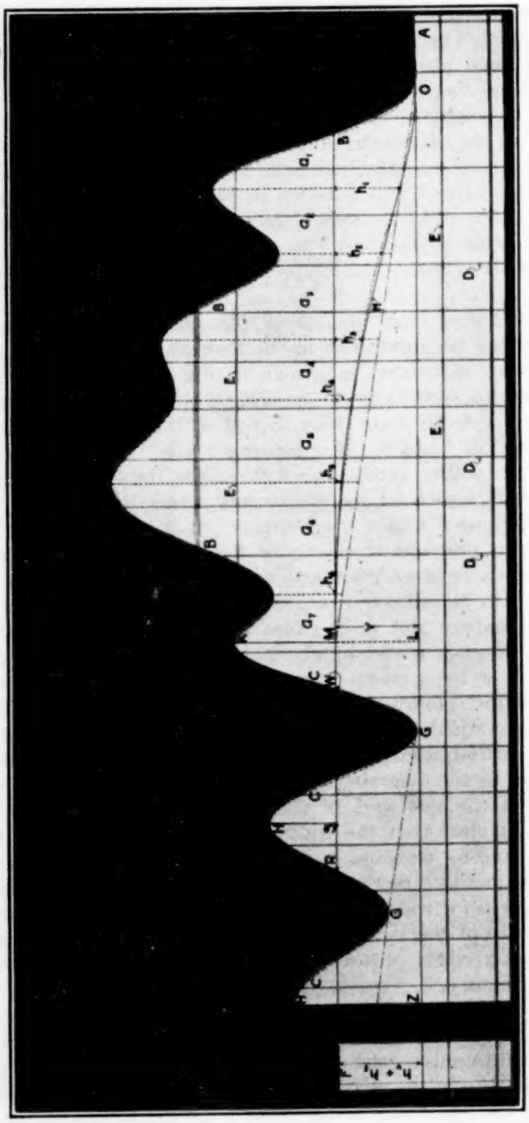

FIG. 8 TYPICAL PRESSURE-TIME DIAGRAM AFTER DELINEATION

ing through the peaks of the waves as from H to H and G to G are equal. The positions of the vertical lines passing through the peaks of the waves at HH, etc., may be determined in several ways, such as by inspection or by the intersection of tangents to the points of inflection of the wave line. Probably the most effective way, however, of fixing the position of the peaks of the waves is by using the relations shown in Fig. 9. The curve in this figure gives the distance RS, measured horizontally along the neutral line [1] about which the oscillations are taking place, from the point of inflection of the wave line to the line passing through the peak of the wave. The proportion of the distance RS to the wave length depends on the amount of damping effect, and the curve is plotted in radian measure with the ratio of wave heights as abscissas and percentage of half wave length as ordinates.

58 4 — The end of the diagram is at the beginning of the first regular wave nearest AA as at HK, and the position of K is determined by measuring back from W the point where the wave passes the neutral line FM,[1] a distance equal to RS. The position of the vertical line through K may also be found by marking off from the peak of the nearest wave a horizontal distance equal to one wave length HH. A vertical line KML is drawn through K intersecting the line F produced at M and the line A produced at L.

59 5 — The point where the pressure line B departs from the line A is marked at O.

60 6 — A straight line is drawn from M to O.

61 7 — A number of vertical lines are drawn from the pressure line BB to intersect the line OM and the areas a_1, a_2, a_3, a_4, a_5, a_6, a_7 enclosed by these lines and corrected for width of slot are measured by planimeter or by scaling in square inches. The sum of these areas will be called A_1.

62 8 — The distance LM, which is a measure of the sum of the friction and velocity heads, is measured in inches and called Y.

63 9 — Trial and error are now used to locate the line ONM which eliminates from the gross area the area produced by the recovery of friction and velocity heads.

64 As a first assumption the straight line from O to M may be taken, but usually it will be found easy to estimate by inspection the approximate position of the line ONM so that only one correction will be necessary.

65 Let $P_1 = \dfrac{a_1}{A_1}$, $\quad P_2 = \dfrac{a_1 + a_2}{A_1}$, $\quad P_3 = \dfrac{a_1 + a_2 + a_3}{A_1}$, etc.,

and let $h_1 = Y(1 - P_1)^2$, $h_2 = Y(1 - P_2)^2$, etc. Having determined the values of h_1, h_2, h_3, etc. from these equations, a point is located on the line separating the areas a_1 and a_2 or this line produced by measuring down vertically from the line FM produced a distance

[1] The neutral line is not always horizontal or coincident with F. See numerical example beginning at Par. 87.

equal to h_1. A distance equal to h_2 is measured down vertically from the line FM produced on the line separating the areas a_2 and a_3. Similarly a distance equal to each of the values h_3, h_4, etc. is measured down from the line FM produced on the respective lines separating the areas a_3, a_4, etc. The points so plotted are then joined together, forming a line from O to M. If the line so plotted fails to coincide with the assumed line, a second trial is made using the new line in place of the one first assumed and proceeding as before. In this manner a base line ONM is found such that when the values of h_1, h_2, etc. are calculated as before but using the new areas a_1, a_2, a_3, etc. above the line ONM, the line joining the ends of h_1, h_2, etc. will coincide with ONM.

66 10 — The area of the diagram delineated by the line $OBBKMN$ is called A and is a measure of the rate of discharge or quantity of water that was flowing in the conduit at the moment the gates began to close. If, however, the final velocity is not zero or relatively small the delineation of the diagram is performed in a slightly different way, as will be explained later (see numerical example, Par. 87 et seq.).

67 In practice it will sometimes be necessary to correct the area A for the effect of special conditions or disturbances such as surges in the forebay, gate well, or surge tank at the entrance to the conduit. This is readily done by observing or recording the height and duration of such surges and subtracting a corresponding area from the area A. Flow measurements may also be made by obtaining two diagrams similar to Fig. 8 simultaneously at different points along the conduit a known distance apart, the difference between the net areas of these diagrams adjusted to the same scale being used in place of the area A. Satisfactory results may also be obtained by using the differential diagram already referred to and illustrated in Fig. 11, which records only the difference in the changes of pressure that occur at two points on the pipe line. In these cases the pipe factor F for that portion of the pipe between the two points only will be used.

Fundamental Principles

68 The method herein described makes use of two well-known principles, the first being Newton's second law of motion, sometimes referred to as the equation of impulse and momentum, and the second (attributed to Joukovsky) being a corollary of the first, namely, the relation between change of pressure and change of velocity of a column of water expressed in terms of the velocity of the pressure waves which are propagated during the change from one end of the column to the other. As the first of these principles is one of the fundamentals of mechanical science, it will be accepted without proof and may be expressed by the equation [1]

[1] See article on Mechanics in Encyclopaedia Britannica, vol. 17.

$$F = M\frac{dv}{dt} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

in which F is a force acting on a mass M and dv/dt is the rate of change of the velocity (acceleration or retardation) of that mass.

69 The second principle is derived from the relation of the first to the phenomena manifested when a change of pressure occurs in an enclosed column of flowing water.

70 When a pressure is applied to the end of a column of water, a time interval t is required for a pressure wave, transmitting the

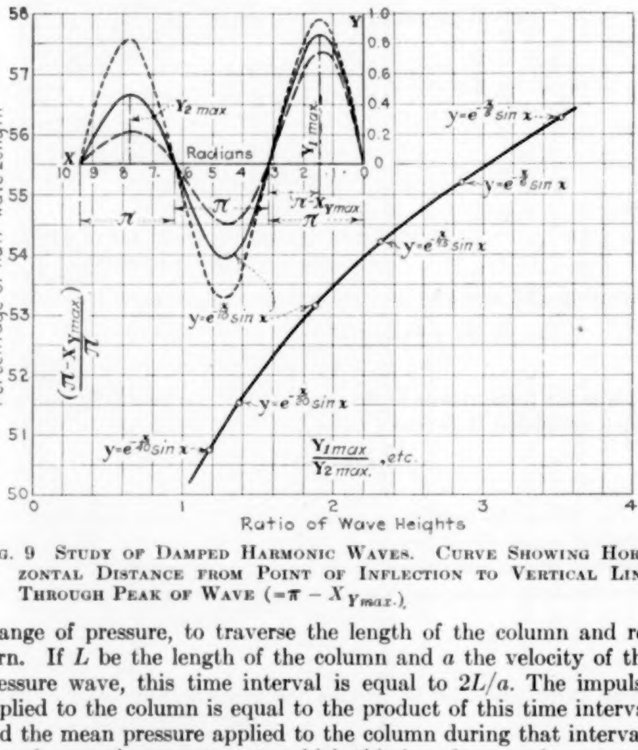

FIG. 9 STUDY OF DAMPED HARMONIC WAVES. CURVE SHOWING HORIZONTAL DISTANCE FROM POINT OF INFLECTION TO VERTICAL LINE THROUGH PEAK OF WAVE ($= \pi - X_{Y max}$).

change of pressure, to traverse the length of the column and return. If L be the length of the column and a the velocity of the pressure wave, this time interval is equal to $2L/a$. The impulse applied to the column is equal to the product of this time interval and the mean pressure applied to the column during that interval. The change of momentum to which this impulse may be equated is the product of the mass of the column and the change of its velocity during the interval in question. The pressure of the water and its velocity are taken at the same instant and quantities of the second order of magnitude are neglected. Let

h_n = pressure in feet above normal existing at the end of any interval of time as defined

h_{n-1} = pressure in feet above normal existing at the end of the interval preceding the interval n

$\frac{2L}{a}$ = a time interval as above defined

L = length of the column of water

A = area of the column of water

w = weight of a cubic foot of water

$\frac{LAw}{g}$ = mass of the water

V_n = velocity of the column of water at the end of the interval n

V_{n-1} = velocity of the column of water at the end of the interval preceding the interval n.

Then

$$\frac{h_n + h_{n-1}}{2} \times Aw = \text{mean pressure applied to the column during the interval } n \quad [2]$$

and

$$\frac{h_n + h_{n-1}}{2} \times Aw \times \frac{2L}{a} = \text{product of this mean pressure and the time interval} \quad [3]$$

also

$$\frac{LAw}{g}(V_{n-1} - V_n) = \text{change of momentum of column of water in question during } n\text{th interval} \quad [4]$$

Equating [3] and [4] and cancelling the common term Aw,

$$\frac{h_n + h_{n-1}}{2} \times \frac{2L}{a} = \frac{L}{g}(V_{n-1} - V_n) \quad [5]$$

This is the form in which this equation will be used later, although when simplified it becomes

$$h_n + h_{n-1} = \frac{a}{g}(V_{n-1} - V_n)$$

71 When the complete change of velocity takes place in a time interval equal to or less than $2L/a$, in which case the resulting pressure is not influenced by returning pressure waves, this relation becomes Joukovsky's equation for maximum water hammer

$$h_{max} = \frac{a}{g} V$$

from which may also be written the general differential equation

$$dh = \frac{a}{g} dV$$

72 In the author's former paper,[1] already referred to, will be found a treatise on the phenomena that occur in a pipe when gradual closure is made of a valve or turbine gates located at the end of a pipe. A thorough knowledge of that subject is of great advantage in understanding the theory and practice of water-flow measurement by the method herein considered.

[1] Pressure in Penstocks Caused by the Gradual Closing of Turbine Gates. Trans. A.S.C.E. 1920.

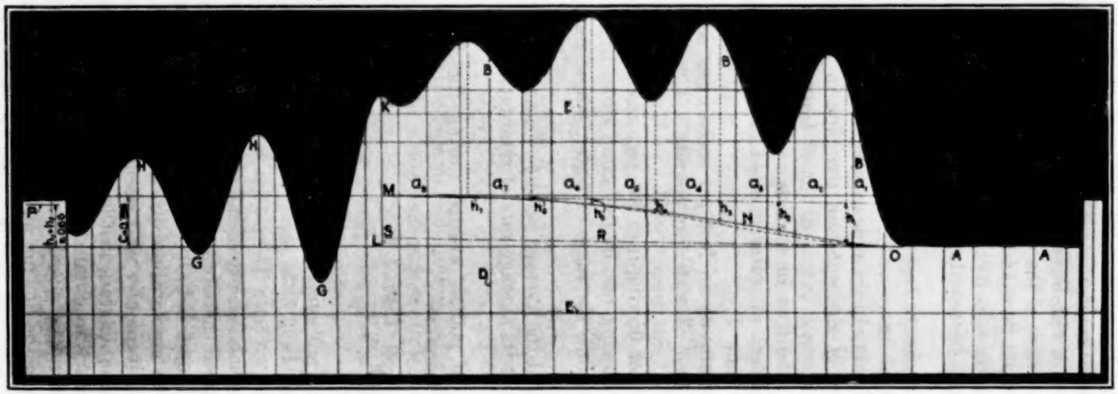

FIG. 10 PRESSURE-TIME DIAGRAM USED IN NUMERICAL EXAMPLE

THEORY

73 Consider a pipe of known length L through which water is flowing at a uniform rate under constant head, its velocity V being fixed by the area of opening of a valve, or the combined area of the passages through turbine gates, located at the lower end of the pipe. Assume that the valve is closed gradually but not necessarily uniformly in the time T, so that the velocity of the water in the pipe is gradually diminished and is finally stopped when the valve is fully closed. During the closure of the valve or gates, as the case may be, the pressure in the pipe rises and after it is closed the pressure falls and rises in periodic waves until damped out by friction. The

rise of pressure during closure of the valve or gates is the manifestation of the force exerted to stop the flow of the mass of water in the pipe. Neglecting for the present the influence of the elasticity of water and of the walls of the pipe, the simple relation determining the magnitude of the force exerted is given above in Equation [1]. If then the mass of the water in the pipe is known and the time taken to stop its flow and the *average* force exerted in doing so are measured, the velocity of the water prior to the interruption of its flow may be readily determined. Let

A = area of the pipe in square feet
L = length of the pipe in feet
Q = quantity of water flowing in the pipe in cubic feet per second
V = velocity of the flow of the water in the pipe in feet per second = Q/A
T = time taken to stop the flow in seconds
P = average pressure in feet of water exerted in stopping the flow during the time T
g = acceleration due to gravity in feet per second per second
w = weight of a cubic foot of water in pounds.

Then if the quantity of water flowing into the pipe during closure is equal to the quantity flowing out in the same time, in other words, if the pipe remains full, the mass of water, whose motion is stopped, equals LAw/g.

74 This mass is flowing at the average velocity V and is brought to rest in the time T, so that its average rate of retardation is V/T ft. per sec. per sec.

75 The average force in pounds exerted during the time T to stop the flow is PwA, therefore

$$PwA = \frac{LAw}{g} \times \frac{V}{T}$$

from which

$$V = \frac{PgT}{L}$$

76 The measurement of the velocity of the water in the conduit by the process herein described therefore requires the determination of the quantities P, g, T, and L. Of these, L is the measured length of the pipe, g is the acceleration due to gravity, and the product of P and T is conveniently and accurately determined by producing a pressure-time diagram in the manner hereinbefore described. It is not necessary to obtain separately the respective values of P and T. It should be noted here that in practice the friction and velocity heads of the flowing water must be accounted for in the manner already explained, and it should be remembered that P is the average, not the maximum, pressure during the time T, and that the influence of the elasticity of water and of the walls of the pipe has been neglected. Whenever this refinement is

necessary the latter may be taken into account and its relation to the velocity destroyed is explained in the author's paper already referred to, which, while not usually essential in the practical use of this method of measurement, will be found useful in fixing a convenient period of time to allow for the duration of the closure of the turbine gates.

77 Referring now to Fig. 8, the ordinates represent pressures to a certain scale and the abscissas represent time to a certain scale. On this or any such diagram let

A = net area in square inches, after eliminating the area produced by the recovery of friction and velocity heads
l = length of diagram in inches from the point of beginning to the point of ending of the closure
r = vertical height in inches corresponding to one foot of pressure change in the conduit
S = horizontal length in inches corresponding to one second of time.

Then the average pressure $P = A/lr$, and the time $T = l/S$, from which $PT = A/rs$. Substituting this value for PT in the equation for V given in Par. 75, then

$$V = \frac{gA}{rSL}$$

and substituting K for g/r,

$$V = \frac{KA}{SL}$$

78 If the pipe were of uniform area throughout its entire length this would be a convenient form in which the final equation could be used. The quantity of water Q flowing in the pipe would then be determined by multiplying the value of V thus found by the cross-sectional area of the pipe.

79 When the pipe is not of uniform area throughout its length but is made up of a series of different sections of lengths l_1, l_2, l_3, etc., having areas A_1, A_2, A_3, etc., respectively, the relations given in Par. 75 hold for each section and the total impulse PT for the combined pipe will be equal to $1/g(l_1v_1 + l_2v_2 + l_3v_3, \text{etc.}) = 1/g\Sigma lv$. As this is the case usually found in practice it will be more convenient to solve directly for Q, which may be done by substituting Q/A_1, Q/A_2, Q/A_3, etc. for V_1, V_2, V_3, etc. The final equation then becomes:

$$Q = \frac{KA}{SF}$$

in which $F = \Sigma(l/a)$, that is, the sum of the quotients obtained by dividing the length of each uniform section of the pipe by the area of that section, and the other symbols are the same as used in Par. 77.

80 Reverting to Pars. 48 and 49, it was shown that the diagram such as Fig. 8 is a record to exact scale of the changes of level of the mercury in tube D and that the change of pressure in the pipe in feet of water is $\frac{1}{12}[M(R+1)-1]$ times the change in level of the surface of the mercury, or, in other words, the ordinate of the diagram measured in inches.

Then r by definition is

$$\frac{1}{\frac{1}{12}[M(R+1)-1]}$$

but

$$K = \frac{g}{r}$$

therefore

$$K = \frac{g}{12}[M(R+1)-1]$$

81 Proof will now be given of the method used in eliminating from the diagram that part of the gross area produced by the recovery of friction and velocity heads.

82 A general proof would require a complete discussion of the matter contained in the author's former paper already referred to, but for the present purpose it will be sufficient to accept the equation given above in Par. 70 and proceed as follows:

83 The second general principle as stated above is

$$dh = \frac{a}{g}\,dv$$

or expressing the relations that exist during any interval as in Equation [5], Par. 70,

$$\frac{h_{n-1}+h_n}{2} \times \frac{2L}{a} = \frac{L}{g}(V_{n-1} - V_n)$$

84 Expressed in words, this means that the excess pressure at the beginning of one "interval" plus the excess pressure at the end of the same "interval" multiplied by the time of one "interval," i.e., $2L/a$, is equal to a constant multiplied by the change of velocity that occurs during that interval. But the first side of this equation is equal to the area of the pressure-time diagram for one interval, if we assume that the curve from the beginning to the end of an interval is substantially the same as the chord between these two points. In practical cases this is always so, but even if extreme cases are considered it will be found on closer study that the differences caused by the departure of the pressure curve from a straight line between interval points are alternately plus and minus and thus tend to equalize the slight error that might otherwise be introduced. Usually the length of an "interval" on a diagram is so small that no difference can be

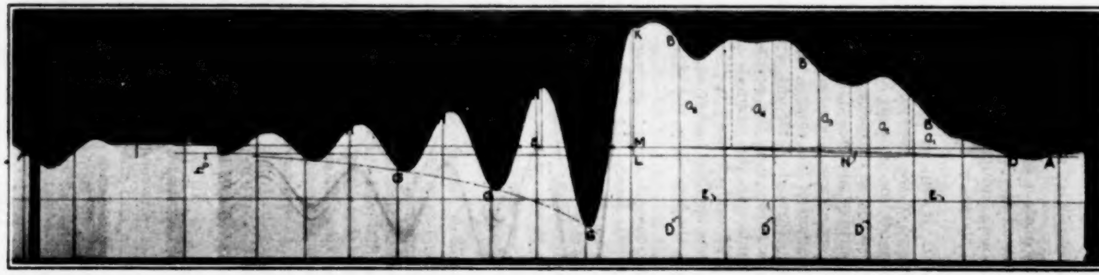

FIG. 11 TYPICAL DIFFERENTIAL PRESSURE-TIME DIAGRAM

found between the area under the curve and the area under the chord.

85 From this it follows that at any time during the closing period the ratio of any part of the net area of a pressure-time diagram to the whole area is equal to the ratio of the velocity destroyed up to that time to the initial velocity. That is, if a_1 is any part of the whole net area A, V the initial velocity, and V_1 the velocity destroyed in the time taken to form the area a_1, then

$$\frac{a_1}{A} = \frac{V_1}{V}$$

86 That part of the gross area of the diagram produced by the recovery of friction and velocity heads may therefore be eliminated as described above, remembering that both friction and velocity heads are proportional to the square of the velocity. That is to say, when half the velocity is destroyed only one-quarter of the friction and velocity heads remain, and so forth.

APPENDIX NO. 1

NUMERICAL EXAMPLE

(Using simple diagram)

87 The foregoing explanation will be made clearer and the use of the diagrams in practical work will be explained by the numerical calculation of the rate of discharge of water flowing in a conduit, using for the example the accompanying data and the diagram shown in Fig. 10. A 20-in. slide rule was used for the computations.

88 Referring to the diagram, Fig. 10, it will be noted that the line through F is slightly higher than the record of the level of the static pressure. In locating and drawing the horizontal line through F it is necessary to consider the level of the water in the forebay at entrance to the penstock at the moment when the gates of the turbine are closed. The record of the static pressure may have been taken at a time when the forebay level was slightly different from its level at the end of the closing time, and this difference must be allowed for. The line through F should represent the static pressure at the moment the gates close or at the point where the pressure-time diagram (during closure) ends. In this example the forebay level, as obtained by the neutral line FM and checked by the record of headwater level taken during the test, was slightly higher at the moment the gates closed than it was at the time the record of the static pressure was taken.

89 In the same way, to obtain the correct height corresponding to $h_v + h_f$ (the sum of the friction and velocity heads) the forebay level just prior to the shutdown must be used. The proper line on the diagram is obtained as before with respect to the level at the time the static pressure is recorded on the diagram. In this example the forebay level just prior to the shutdown was 0.23 ft. lower than it was at the time the record of static pressure was taken.

90 Then changing from feet of water to inches on the diagram,

$$0.23 \div \frac{[M(R+1)-1]}{12} = 0.073 \text{ in. on the diagram}$$

The correct height of $h_v + h_f$ under running conditions is thus obtained by measuring down from the static-pressure line the distance corresponding to the change in forebay level, i.e., 0.073 in. as at PP'.

91 Instead of producing horizontally through F the line representing the forebay level at the end of the closing time, a line representing the surge in the forebay during the time of closure may be plotted on the diagram separately, thus forming a rising datum as indicated by the dotted line ORS, and should not be included in the ordinate C. The true value of C representing $h_v + h_f$ only or the ordinate MS is then pointed off at intervals above and along the line ORS and a line is then drawn through these points parallel to ORS.

92 In this case, however, where the change in forebay level was small the position of the recovery line ONM was the same when calculated either by assuming the whole ordinate LM was recovered or by the more precise method of plotting a rising datum and the recovery of the true ordinate MS representing friction and velocity head.

93 The first assumption for the curve of rising gradient between O and M, shown on the diagram by dotted line, having been made, and the diagram divided into small areas a_1, a_2, etc., these small areas, above the dotted line OM, are now measured by planimeter and added together to give the total area, the various steps being taken in the following manner and the results set down as shown in Table 1.

94 As already stated, the method herein described measures only the quantity of water actually shut off, and if any quantity remains flowing

in the conduit it must be measured separately and its amount added to obtain the total discharge. If this leakage is relatively small, no appreciable error is introduced by assuming that the leakage velocity is zero when calculating the results from the diagrams, but since the difference between the squares of two quantities is not the same as the square of their difference, it becomes necessary to have regard to this fact when the leakage quantity is relatively large. As the complexity or length of the calculations is not materially increased by taking leakage velocity into account in calculating the results from the diagrams, it is perhaps better to do so at all times whether the amount be large or small, and in the following example this will be done.

95 At the top of Table 1 on the left-hand side, the name of the plant, the date of the test, the number of the unit, and the number of the run will be entered for identification, and on the right-hand side, the pipe factor F, the instrument constant K, the pressure scale of the diagram, and the value of the ordinate $h_v + h_f$, all of which have already been explained, will be set down for convenient reference.

96 The first four columns of the table show respectively the designating letters of the small areas, the planimeter readings, the differences between these readings, and the net area of each small division. The mean of at least two determinations of each area is used in order to do the planimeter work as precisely as necessary. The planimeter used in this example was one of the "Precision" type which reads to half-scale, so that the sum of two separate determinations would therefore give the mean area to full scale. It will be noted that in the fourth column the correction for slot has been made by multiplying the vertical height of that part of the area which requires the correction by the width of slot (in this case 1/32 in.) and subtracting this product from the gross area, as explained in Par. 54. This is simpler than marking on the diagram the dotted line as shown in Fig. 8.

97 The fifth column in the table is obtained by adding together successively the net areas in column 4. The last line in this column will show the total net area of the diagram above the assumed recovery line. In the first trial it is not necessary to check the sum of the small areas by measuring the total area direct, but this is done only after the final-recovery line has been located.

98 An imaginary area is now added to the total area in column 5 proportional to the leakage velocity which has not been destroyed. That is to say, if the leakage velocity had been destroyed, the area A would have been proportionately larger and the increase in area would have occurred after the point K (Fig. 10) had been reached. The amount of the additional area is easily calculated because the leakage quantity having been determined (in this case 10.7) as explained in Pars. 22 and 23, the equation $A_L = qSF/K$ which is derived from the final equation given in Par. 79 may be used. In this equation q is the leakage quantity, A_L the imaginary leakage area, and the other symbols have the same significance as before.

99 It is now necessary to determine the value of S, which is done simply by measuring the distance in inches between an even number of second marks to be taken as nearly as possible within the length of the main part of the diagram. As many seconds as possible should be chosen. In the example the distance marking 18 sec. was 8.505 in. and therefore $S = 8.505/18 = 0.4723$. Inserting this value in the above equation, A_L is found equal to $(10.7 \times 0.4723 \times 1.939)/96.55 = 0.102$.

100 This value having been added to the total at the foot of column 5, the remainder of the table may be completed as follows:

101 In column 6 the ratios r_1, r_2, r_3, etc., of the successive sums of the small areas to the total area $A_1 + A_L$ are set down. Thus for example, on the first line the ratio will be $0.745/16.326 = 0.046$. In column 7 the value of $(1 - r)$ and in column 8 the value of $(1 - r)^2$ are computed and entered in their respective places. Column 9 shows the product of the amounts shown in column 8 and the value of $h_v + h_f$. As previously ex-

plained, column 8 may be multiplied by the value of C, which includes $h_v + h_f$ and the surge in the forebay if this is small. In this case the values in column 9 would be measured down from the line FM produced to locate the recovery line ONM. But when the forebay surge is plotted on the diagram as described in Par. 91, the value to be used is only the height of the ordinate $h_v + h_f$, or what is the same thing, the ordinate MS. In this example the latter value has been used and as shown is equal to 0.66. On the diagram, Fig. 10, C is shown equal to 0.798, but the difference between the two values must be kept clearly in mind.

102 The figures in column 9 represent the values of the ordinates h_1, h_2, h_3, etc., to be measured down from the line $P'M$, which is parallel to ORS. The points thus determined are then joined together to form the recovery line ONM. If the line ONM so formed is not relatively close to the trial line just assumed it will be necessary to repeat the process explained above, using the line ONM as a new trial line. Usually, however, after a little practice it will be found easy to locate the first trial line so that a second trial need not be made, and the line ONM found from the first set of computations will be the true recovery line as closely as it can be drawn.

103 The recovery line ONM having thus been found and drawn, the total area above this line and within the limits of the diagram $OBBKMN$ as previously defined is then measured by planimeter, the total area being corrected for width of slot as before. In this example the area found in this way was 16.406 sq. in. and the total correction for slot 0.229 sq. in., making a total net area of 16.177 sq. in.

104 A slight correction of the quantity A/S which appears in the final equation is sometimes required to allow for the shrinkage or stretching of the blueprint, and as the photographic films have no appreciable variation in dimensions before and after developing, the correction may be made by taking the ratio of width of film to the corresponding width of print; and as the value of S is obtained from the print itself, the correction may be applied directly to the value of A. The width of film, measured in fifths of an inch, was 54.58, and the corresponding width of print, 54.47. The corrected net area of the diagram is therefore $54.58/54.47 \times 16.177 = 16.21$ sq. in.

105 The equation $Q = KA/SF$, given in Par. 79, may now be solved from the data already obtained as follows:

$$Q = \frac{96.55 \times 16.21}{0.4723 \times 1.939} = 1708$$

and adding the leakage $q = \underline{\quad 10.7}$

the total discharge $= 1718.7$ cu. ft. per sec.

106 The remaining figures in the data sheet, Table 1, will require but little explanation as their significance is sufficiently obvious. The value of the friction head h_f in the penstock to the point where the apparatus was attached may be found by separating out the value of the velocity head h_v from the sum of $h_v + h_f$. The latter is equal to 0.66 in. on the diagram, which corresponds to 0.66×3 ft. of water pressure. The net head at entrance to the turbine casing will be determined in the usual way, and from the records of headwater and tailwater the total friction loss to this point will be obtained. From the power records, generator efficiency, net head, and quantity of water, the combined efficiency of turbine and generator, and the turbine efficiency are derived.

TABLE 1 TURBINE EFFICIENCY TESTS GIBSON METHOD

Plant: Hydraulic
Date of test: Aug. 17, 1920
Unit No. 16
Run No. 15
Planimeter readings half scale

$F = \Sigma \frac{lv}{q} = 1.939$
$K = 96.55$
1 in. on diagram = 3.00 ft.
$h_v + h_f = 0.66$ in.

(1) No.	(2) Planimeter readings	(3) Diff.	(4) Mean area corrected for slot	(5) Σa	(6) r	(7) $(1 - r)$	(8) $(1 - r)^2$	(9) $(1 - r)^2 \times (h_v + h_f)$
a_1	46.131 45.722 45.307	0.409 0.415	0.824 0.079 0.745	0.745	0.046	0.954	0.912	0.602
a_2	48.374 47.186 45.995	1.188 1.191	2.379 0.015 2.364	3.109	0.191	0.809	0.655	0.432
a_3	50.056 49.189 48.322	0.867 0.867	1.734 0.058 1.676	4.785	0.293	0.707	0.500	0.330
a_4	2.621 1.345 0.068	1.276 1.277	2.553 0.006 2.547	7.332	0.449	0.551	0.304	0.202
a_5	4.823 3.703 2.587	1.120 1.116	2.236 0.041 2.195	9.527	0.584	0.416	0.173	0.114
a_6	6.964 5.818 4.677	1.146 1.141	2.287 .000 2.287	11.814	0.724	0.276	0.076	0.050
a_7	8.668 7.672 6.681	0.996 0.991	1.987 0.024 1.963	13.777	0.844	0.156	0.024	0.016
a_8	10.657 9.429 8.204	1.228 1.225	2.453 0.006 2.447	16.224 = A_1	.	.....	.....	.....

Leakage area A_L = 0.102
Total........ 16.326

Leakage area on diagram:

$$\frac{10.7 \times 1.939 \times 0.4723}{96.55} = 0.1016$$

Correction for print $= \frac{54.58}{54.47} \times 16.177 = 16.21$

Total net area:

40.643	8.210	
32.433	8.196	...16.406
24.237		0.229
		16.177

Net area A = 16.21
$S = 8.505/18 = 0.4723$

$$Q = \frac{96.55 \times 16.21}{1.939 \times 0.4723} = 1708$$

Leakage q = 10.7
Total cu. ft. per sec. = 1718.7

$h_v + h_f = 3.00 \times 0.66 = 1.98$

$h_v = \left(\frac{1718.7}{188.6}\right)^2 \div 64.4 = 1.29$

To apparatus h_f = 0.69
Headwater elevation = 556.20
Tailwater elevation = 342.15
Gross head = 214.05
(To turbine) h_f = 3.15
Net head = 210.90

Generator kw. = 27,648
Generator hp. = 37,061
Turbine hp. = 37,600

Combined efficiency = 90.3 per cent
Generator efficiency = 98.55 per cent
Turbine efficiency = 91.6 per cent

APPENDIX NO. 2

THE CORNELL TESTS

107 In order to obtain satisfactory proof of the accuracy of this new method of measurement, arrangements were made with Dean E. E. Haskell, of the College of Civil Engineering at Cornell University, to have a thorough test made by comparison with volumetric measurements at the hydraulic laboratory of the university. This experimental work was undertaken in the interest of The Niagara Falls Power Company in order to comply with the requirements of the engineers of the United States War Department in charge of the regulation of the use of the waters of the Niagara River under the treaty between the United States and Great Britain.

108 The laboratory work was carried out under the direction of Dean Haskell and under the personal supervision of Dr. E. W. Schoder, the professor in charge of the laboratory. Dr. Schoder, with the help of his assistants, Messrs. J. H. Stalker and J. A. Thomas, made the volumetric measurements, and the author and his assistant, Mr. P. F. Kruse, had charge of the Gibson measurements.

109 Dean Haskell, Dr. Schoder, and the author submitted separate reports covering the experimental work in detail, but for the purpose of this paper it has been considered necessary to include only the summary of the results as compiled in Table 2, which gives the comparison of the volumetric and Gibson measurements. An examination of this table will show the precision that has been attained in the measurement of water by the use of the method and apparatus described in this paper.

TABLE 2 COMPARISON OF VOLUMETRIC AND GIBSON MEASUREMENTS

(Record of tests made under approximately the Niagara conditions)

Test No.	Field No.	Date 1920	Discharge, cu. ft. per sec. Volumetric	Gibson	Per cent variation	Remarks
20-E-1	45	June 11	20.27	20.31	+0.2	Tube R = 1.335
20-E-2	46	June 11	20.42	20.52	+0.5	
20-E-3	47	June 11	20.45	20.65	+0.9	
			20.38	20.48	+0.5	Mean of series.
30-A-1	5	May 24	30.38	30.51	+0.4	
30-A-2	6	May 25	30.63	.....	.....	Poor diagram (not completed).
30-A-3	7	May 25	30.69	31.15	+1.5	
			30.57	30.82	+0.8	Mean of series.
40-A-1	8	May 25	42.24	42.17	−0.2	
40-A-2	9	May 25	41.94	42.28	+0.8	
40-A-3	10	May 25	42.22	42.09	−0.3	
40-A-4	12	May 25	.....	41.96	.....	Volumetric not taken.
			42.13	42.12	0.0	Mean of series.
40-C-1	30	June 4	40.36	40.41	+0.1	
40-C-2	31	June 4	40.36	40.90	+1.3	
40-C-3	32	June 5	40.98	40.88	−0.2	
			40.57	40.73	+0.4	Mean of series.
50-A-1	15	May 29	46.82	.....	.....	Film fogged, diagram invisible.
50-A-2	16	May 29	46.95	46.60	−0.7	
50-A-3	17	May 29	46.98	46.60	−0.8	
			46.92	46.60	−0.7	Mean of series.
50-C-1	33	June 7	50.30	49.95	−0.7	
50-C-2	34	June 9	50.66	51.47	+1.6	
50-C-3	35	June 10	50.93	51.15	+0.4	
			50.63	50.85	+0.4	Mean of series.
Total of Means......			231.20	231.60	+0.2	Mean variation of all tests.

DISCUSSION [1]

Clemens Herschel.[2] Water measurements, as described in the papers by Messrs. Allen and Taylor and Mr. Gibson, are essentially methods of work, to be used by engineers, to arrive at certain desired results. The principal use that can be made of these results in the existing trend of hydraulic engineering is to find the efficiency of hydraulic turbines at disjointed intervals. Neither of the methods described is hardly competent for more than this.

Neither method will record continuously on a diagram the discharge of a hydraulic turbine. But precisely this thing has been done at the Ogden (Utah) power house of the Utah Light and Power Company for twenty years, providing assurance to the operating force that the water supply was constantly being used to the best advantage, and that all parts of the turbine were in normal condition. In one case we have engineers using able methods to make single detached gagings. In the other case a machine meters the water continuously, records its flow on a diagram, and keeps a constant watch on the condition of the turbine. This illustrates the difference between the use of an engineer's instrument and a water meter. And if metering be desired, it is believed that for hydraulic-turbine practice the venturi meter stands alone.

The fact that the venturi meter uses a coefficient, should cause no hesitancy in its use. It is now thirty-five years since it was first rated, and a large body of experimental determinations of this coefficient of rating are on record.[3] Some engineering determinations may be accepted as proved. The venturi-meter coefficient is one of these. Such experimental coefficients have never been found to vary materially in the case of pipes and penstocks of the ordinary sizes. For example, on the New York water supply there are meters capable of passing 800 cu. ft. per second. The sum of the records of a dozen or twenty branch meters drawing out a quantity of water passed by the master meter agrees with the record of the latter.

The use of a venturi meter in any hydraulic test permits the quantity of discharge to be selected and attained before the experiment begins. Thereafter, this quantity can be varied by even increments, an advantage in making such tests that is very great. Often a temporary meter tube could be put into the penstock, upstream or downstream from the turbine. Carpenter work and machinist work is as legitimate an expense in testing water wheels as any other.

Frequently, continuous metering of the turbine discharge is desirable. There is no necessity for a lapse of two or three days to

[1] Joint discussion of Papers Nos. 1902 and 1903.
[2] Hydraulic Engineer, New York, N. Y.
[3] Trans. A.S.M.E., vol. 42, p. 191.

exist before the data and results for a certain discharge are ascertained. Loss of head need never be of material amount, and, if desired, could be entirely annulled by the use of one or two small bypasses.

Consideration thus should be given in all cases whether it will be better to meter continuously, rather than merely to measure at disjointed times. From the number of power plants that are putting permanent venturi meters in the line of their penstocks, the writer infers that metering often is considered the better choice.

H. K. Barrows.[1] The salt velocity method as described by Messrs. Allen and Taylor appears to be analogous to that of float measurements, in that the elements involved are the area and length of channel used and the time required for the passage of the charge of salt solution between the two electrodes. This method has the advantage that the volume of the conduit can be determined with great accuracy. The limitation of accuracy of the method appears to be in the observation of the time of salt passage. This will vary somewhat, and the resulting accuracy will be less where the distance between electrode sections is short and the time of passage also short. In such cases as those of the tests at the Laurentide Power Company, where the distance was only about 55 ft. and the time of passage 10 seconds, a relatively small error in time will make a considerable percentage error in the result. Under such conditions the salt solution should be introduced some distance above the first electrode.

In regard to the laboratory tests where the results are referred to the discharge over a Francis weir as shown in Fig. 6 of the Allen and Taylor paper, the authors apparently used the mean Francis coefficient of 3.33. It should be remembered that this coefficient is likely to vary at least one per cent from the mean, and that the weir thus is not an infallible standard of comparison. On the other hand, the tests with the 8-in. pipe show a high degree of accuracy as compared to the discharge determined by weighing. The authors seem to have thoroughly demonstrated the accuracy of the salt velocity method in general.

In the commercial testing of water turbines by means of specially constructed weirs, or by the chemical method of using salt solution, the expense is usually prohibitive. Current-meter methods are usually impractical, due to condition of flow. The salt velocity method would appear to be one not involving great expense, although a statement regarding relative costs would be of interest.

This new method of measuring discharge is especially timely, applying, as it does, to the testing of water turbines with closed-flume or penstock settings. While this has often been accomplished in the past for penstock settings by the use of the pitometer, the new method appears to be simpler and more accurate and

[1] Consulting Engineer, Boston, Mass. Discussion refers to Paper No. 1902.

adapted as well to concrete-approach flumes where the former method is impracticable. As concrete-approach flumes and vertical single-runner units have now become quite standard for low- and medium-head developments, the new method should have a wide field of use.

JOHN S. RIDDILE.[1] The authors of the paper on the salt velocity

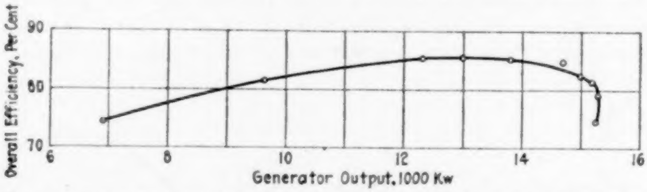

FIG. 1 RELATION OF GENERATOR OUTPUT AND OVERALL EFFICIENCY ON 22,000-HP. HYDROELECTRIC GENERATOR OF LAURENTIDE POWER COMPANY, LTD., ORDINARY SCALE

method of water measurement carried out the greater part of their work in the laboratory, but on such a scale and in such manner that there was good reason to believe that the use of this method on a large scale would have results of the same order, and serve to demonstrate its usefulness in determining water-wheel efficiency.

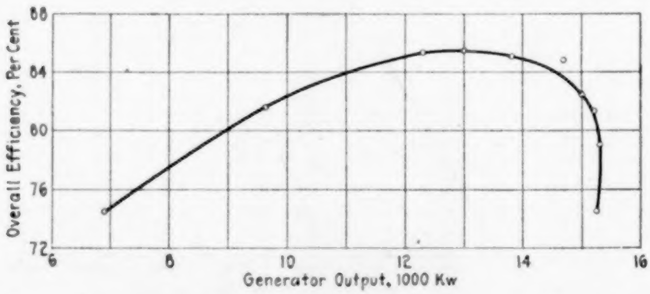

FIG. 2 RELATION OF GENERATOR OUTPUT AND OVERALL EFFICIENCY ON 22,000-HP. HYDROELECTRIC GENERATOR OF LAURENTIDE POWER COMPANY, LTD., ENLARGED SCALE

The Laurentide Power Company happened to have conditions affording an opportunity for such a practical trial.

The conditions at the power plant were not especially simple, and it was well understood that the laboratory experiments should be supplemented by investigation at the plant. The water wheel is rated at 22,000 hp. at 84 ft. head and the water is delivered by the penstock at a rate greater than 2500 cu. ft. per sec. It passes

[1] Manager of Power, Laurentide Power Company, Ltd. Grand Mere, Quebec, Canada. Discussion refers to Paper No. 1902.

through a short, rectangular, converging penstock, as shown in Fig. 15 of the paper.

While the electrodes of a screen or grill design give exceedingly accurate results, equally satisfactory results may be obtained by placing long, vertical electrodes in one plane at two vertical sections of the penstock. The volume of the penstock between these two planes was accurately measured.

The results of an efficiency test on the unit are best shown by the accompanying curves, Figs. 1 to 3. Figs. 1 and 2 show the relation between generator output and overall efficiency of the unit.

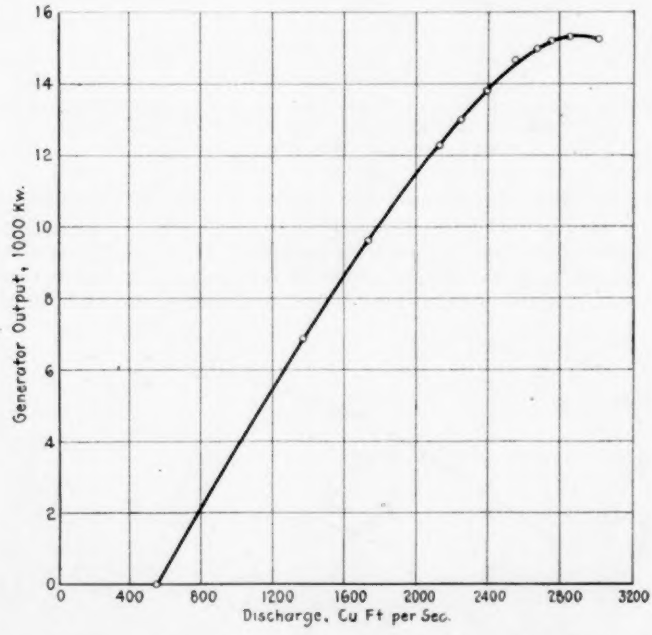

FIG. 3 RELATION OF GENERATOR OUTPUT AND DISCHARGE ON 22,000-HP. HYDROELECTRIC GENERATOR OF LAURENTIDE POWER COMPANY, LTD.

Fig. 1 is drawn to a usual scale and indicates test points all lying very close to a smooth line. Fig. 2 is drawn to a more open scale to emphasize deviations of the test points from the line, the maximum deviation being about one per cent. Fig. 3 shows generator output and wheel discharge. While all points are close to some line, this fact does not establish the accuracy of the results, but does indicate consistency.

A portion of the investigation carried out by the authors to demonstrate the accuracy of the method consisted of an exploration

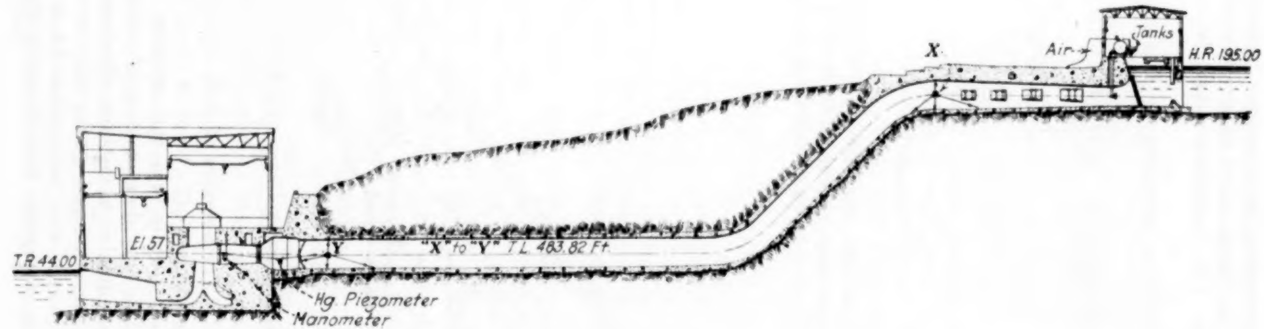

FIG. 4 ARRANGEMENT OF NO. 6 UNIT AT SHAWINIGAN FALLS, SHOWING LOCATION OF ELECTRODES, ETC.

across the penstock by a method of traverses, as described in Pars. 102 to 114. The apparently conclusive results of this study were confirmed in a most substantial way by the authors during a field test on one of the units of the Shawinigan Water and Power Company, the data of which the writer is able to present through the kindness of Mr. Julian C. Smith.

The unit at Shawinigan Falls is supplied through a 20-ft. concrete penstock of unvarying circular section, some 500 ft. long, as shown in Fig. 4. The intake to the penstock, called a gathering tube, consists of four parallel tunnels feeding at an angle into an elliptical diverging extension of the penstock as shown in Fig. 5. Each tunnel converges in section; the lengths vary respectively from about 10 ft. to 67 ft.

Since the other work of the authors left no room for doubt that their measurements on the long penstock would be accurate, a simultaneous test of the water passing through the five portions of the gathering tube could be compared with the discharge of that same water through the long penstock. The results of a series of such check tests are shown in Table 1. The quantities and velocities are similar to those in the 22,000-hp. unit. The maximum deviation was 0.7

per cent. These results of measurement on a complex system of tubes, averaging some 50 ft. in length, gave us any needed confidence in the test made on the Laurentide single 55-ft. penstock.

The means for use of this method as developed by Professor

TABLE 1 COMPARISONS OF DISCHARGE AS MEASURED IN GATHERING TUBE AND MAIN PENSTOCK

Cubic Feet per Second — Gathering Tube (A)	Cubic Feet per Second — Main Penstock (B)		Differences, Per cent of (B)
2904	2924		−0.7
2655	2651		+0.2
2618	2604		+0.5
1884	1891		−0.4
1874	1882		−0.4
1595	1584		+0.7
		Average	0.0

Allen appeared well calculated to provide for accuracy. A single test at a given rate of discharge was made in a small fraction of a minute. Tests were repeated easily, and for each load several tests

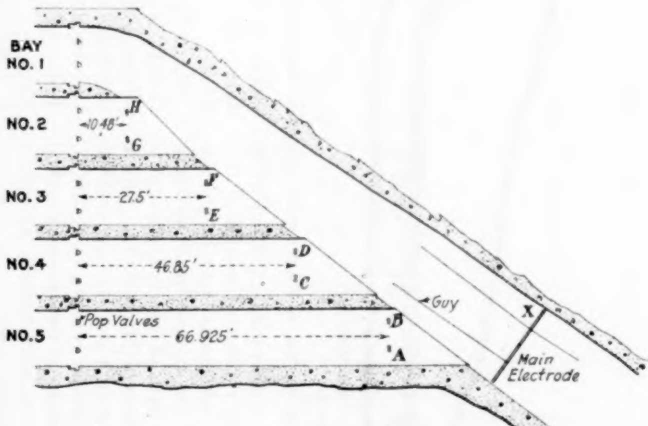

FIG. 5 ARRANGEMENT OF ELECTRODES IN GATHERING TUBES

were made, all in a few minutes. The averages of such repeated tests appear as test points on the curves.

One convenient and important feature was the accuracy of measuring generator output. This was done with a laboratory watt-hour meter read for an exact number of minutes, which gave a true average value of the power developed during that period when the several water measurements were being made for each load. The chances for error due to possible fluctuations in com-

mercial loads were thus definitely reduced. The net results of all this work assured us that the efficiency test had a highly satisfactory degree of accuracy.

R. W. Angus. The most valuable application of both the salt velocity and the Gibson methods is in measuring the efficiency of hydroelectric units. The methods appear to differ in the following respects: In the Gibson method the measurement extends over a very short time, and as efficiency depends on both the water and the electrical measurements, both must be made with accuracy if the computed efficiency is to have any real meaning. It is doubtful if the electrical-output measurements have the same accuracy as the water measurements made by the Gibson method, partly because the electrical changes are not in phase with the hydraulic ones. Hence, if the load is not absolutely steady, and the speed is increasing, for example, the water measured may be storing up kinetic energy in the rotating elements as well as delivering power, so that the output would show a lower efficiency than that actually obtained.

On the other hand, the salt velocity method described by Professor Allen permits the average of the electrical results to be used, since water measurements are being made that can be continued over a considerable period of time. Even though the electrical measurements should not be made with absolute accuracy, or the speed of the unit should vary slightly, the average over a period, as taken by a wattmeter, might give closer results in regard to efficiency.

Professor Allen in Par. 32[1] gives a coefficient of 3.33 for the Francis weir. It is not clear whether Professor Allen intends this particular coefficient to be used or not. The coefficient from the Francis experiments seem to vary from 3.30 to 3.36, with a possibility of 0.3 per cent error.

Both the Gibson and the salt velocity methods can be used with accuracy on long penstocks. There are many plants, however, especially low-head plants, where the distance from headwater to turbine is very short and in such circumstances one method may be more accurate than the other. While Mr. Gibson has not given any figures for such plants, the experiments at the Laurentide Power Company apparently show that the salt velocity method is equally applicable to low-head and high-head plants.

Paul F. Kruse.[2] The writer has been deeply impressed by the fact that throughout the application of the Gibson method to widely different conditions of pipe-line design and various types of governors, turbine gates, or penstock valves, consistent and accu-

[1] Paper No. 1902.

[2] Sanderson & Porter, New York, N. Y. Discussion refers to Paper No. 1903.

rate results have been obtained which have invariably verified the theoretical relations employed by the author.

In connection with Pars. 16 and 18, the writer wishes to call attention to the practical relation between gate travel and the characteristic shapes of the pressure-time diagrams resulting from different heads and different types of turbine gates and penstock valves. While these factors do not affect the area of the pressure-time diagram from which the velocity of flow is computed, the writer has found a prior consideration of them useful in choosing the time and rate of closure, and in the adjustments of the Gibson apparatus, in order to obtain the most easily and accurately interpreted diagram.

It is interesting to note the similarity in the shapes of the pressure-time diagrams in the practical application of the Gibson method, and the theoretical shapes of the rise of pressure curves as affected by the head. For low heads and uniform reduction of gate openings the pressure-time curve rises gradually to a maximum at or near the end of the closure. For moderate to high heads the characteristic curve shows the pressure to rise rapidly at the beginning of the stroke and thence to be maintained fairly steady to the end of closure. The pressure-time diagram, Fig. 10 of Mr. Gibson's paper, for a head of approximately 215 ft., is a practical example of the latter type.

The characteristic shapes of the diagrams are somewhat affected by the type of gate or valve as well as by the head. With wicket gates a uniform motion of the gates produces a uniform reduction of the discharge opening. Cylinder gates destroy the major part of the velocity in the latter part of the stroke, causing a maximum pressure ordinate at or near the end of the diagram. A butterfly valve in the penstock gives a characteristic shape of diagram for uniform motion of the gate, in which the pressure rise is a maximum at a point approximately intermediate between the beginning and end of the stroke, dropping back to approximately normal pressure at or near the end of closure. Due to these differences it may be desirable to instruct the operator at the governor or valve to vary the rate of gate travel in the manner predetermined as best suited to the particular case.

The equations employed in the Gibson method are composed of factors all of the first order and entirely free from experimental coefficients. The accuracy of water-flow measurements by this method depends only on the precision with which the physical measurements of pipe lines, calibration of apparatus, and delineation and measurements of the diagrams are made. It may be expected, therefore, that flow measurements may be made in the field with laboratory accuracy.

E. B. Strowger.[1] The writer has made practical use of the Gibson method in tests of various water wheels in the plants of the

[1] The Niagara Power Co., Niagara Falls, N. Y. Discussion refers to Paper No. 1903.

Niagara Falls Power Company. As an indication of the accuracy of water measurements by this method two examples may be cited.

The first case was the testing of a 55,000-hp. turbine under a head of 312 ft., the penstock to the turbine being about 500 ft. long. Two independent measurements of discharge were made, with separate instruments connected at different points on the penstock. The lengths of penstock to the first and second instruments were 369.17 ft. and 309.88 ft., respectively. The diameters of penstock at the first and second instruments were 16 ft. and 14 ft., respectively. Pressure-time diagrams were taken simultaneously, with

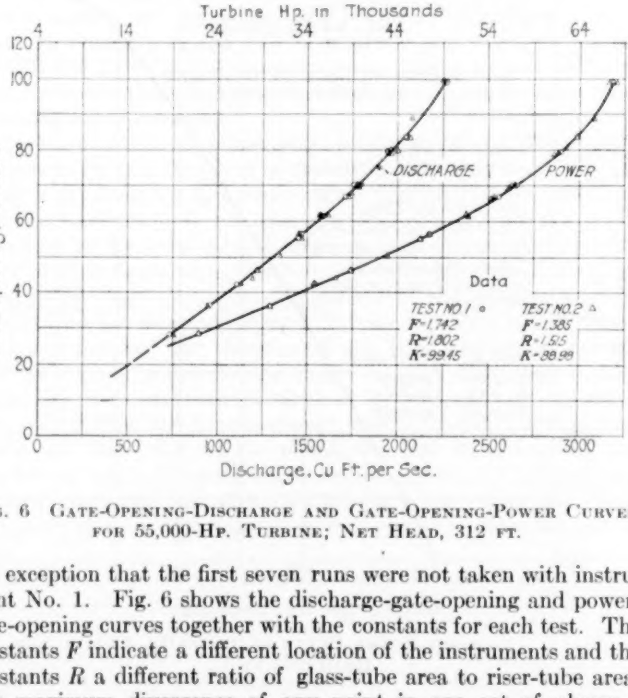

FIG. 6 GATE-OPENING-DISCHARGE AND GATE-OPENING-POWER CURVES FOR 55,000-HP. TURBINE; NET HEAD, 312 FT.

the exception that the first seven runs were not taken with instrument No. 1. Fig. 6 shows the discharge-gate-opening and power-gate-opening curves together with the constants for each test. The constants F indicate a different location of the instruments and the constants R a different ratio of glass-tube area to riser-tube area. The maximum divergence of any point in one set of observations is about 0.5 per cent and in the other set of observations about 1 per cent. Two sets of power determinations are shown in Fig. 6. The power output was measured by the same instruments in both tests, but a slight variation in head observations caused a slight difference in correcting the power readings to a uniform head.

The second test cited is that of a 10,000-hp. unit operating under a head of 212 ft. with a penstock to the turbine 250 ft. long and 9 ft. in diameter. Two tests were made about two months

apart with independent apparatus for each test as noted in Fig. 7. The area of the penstock at the point of attachment in test No. 1 was 50 sq. ft. and in test No. 2 was 57.4 sq. ft. The lengths of penstock from forebay to point of connection in tests Nos. 1 and 2 were 256.20 ft. and 237.66 ft., respectively. Fig. 7 shows the power-discharge curve established by the two tests, the results being practically identical.

The ability to make repeated tests on the same unit with identi-

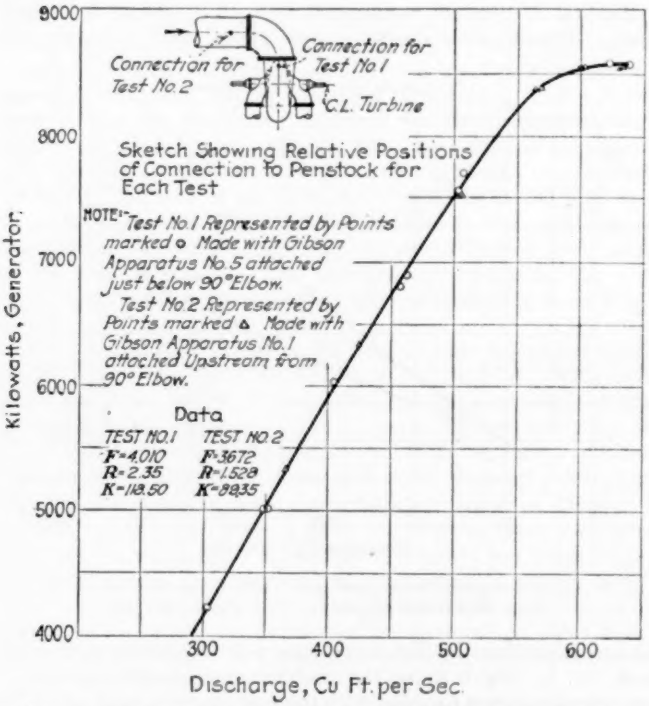

FIG. 7 DISCHARGE-POWER CURVE OF 10,000-HP. UNIT; GROSS HEAD, 212 FT.

cal results, using independent equipment, not only serves as an indirect check on the work, but indicates that a water wheel itself when once calibrated is an indicating flowmeter. It is only necessary to establish the power-discharge or the gate-opening-discharge relation by a test which may be made by the Gibson method. It may be desirable to recalibrate every two or three years or after major repairs on the wheel.

R. D. JOHNSON. The two methods that have been presented in the foregoing papers offer probably the most accurate means for

measuring water known at the present time. Both are direct methods in the sense that they do not depend upon experimental coefficients. Professor Allen has divided water measurements into two classes, a direct method analogous to weighing or measuring by volume, and an indirect method which depends upon experimental coefficients. The two methods which have been presented are direct methods and depend upon simple fundamental laws, and their accuracy is merely the accuracy of the observer. Both are just as accurate, intrinsically, as weighing or measuring the water. The Gibson method is a gravimetric method. The momentum is weighed and the volume is known. In principle the method is the same as running the water into a pail and then weighing it. Professor Allen's method, on the other hand, is purely volumetric. Using the same comparison, the pail of water is not weighed, but the water is run into a pail of known volume. Both methods, in principle, are much alike and there is nothing to choose between them in the matter of accuracy.

This statement is made advisedly as the result of a personal experience in 1914 at the plant of the Salmon River Power Company where a great many measurements of the water in a 10,000-ft. pipe line were made by the color velocity method. This pipe line was regulated by a differential surge tank which also measured the water in three ways when the flow was interrupted by closing the wheel gates; one way is by application of the impulse diagram derived from the measured surge with respect to time with a known volume in the pipe line; the second way is to take account of the total volume of water flowing into the tank following a shutdown and to compute the work done by it against the pressure, thus determining the kinetic energy destroyed and thereby computing the flow prior to the shutdown; the third method, the simplest of all, is to plot a curve indicating the diminishing rate of rise in the surge tank of known area, and to project this curve to its origin where the tangent at that point measures the rate of flow.

The purpose of these elaborate water measurements was not to demonstrate the usefulness of the surge tank as a meter but rather for the purpose of calibrating the wheel gates by the color velocity method, which was thought to be quite reliable. In fact, the accuracy of the water measurements by the surge tank or even its function in this respect was not thought of until Mr. Gibson demonstrated his method.

The surges themselves, however, were carefully compared by the above three methods with the flow as found by the color velocity method and a perfect check established to agree almost beyond belief.

It is therefore now apparent in the light of Mr. Gibson's explanation of his method that the flow was actually measured by the surge tank, and that these measurements constituted a perfect check upon the color velocity method, which latter does not differ

in principle from the Allen method. It merely differs in the kind of eyes which are used to determine the passing of the particles suspended in the water, and I think that Professor Allen has demonstrated that the manner by which he sees them is more universally applicable and more accurate in some cases than the color velocity method. On the other hand, it is also evident that at least one of the methods by which the water was measured in the surge tank agrees perfectly in principle with Mr. Gibson's method: It therefore appears, as above stated, that there is nothing to choose in the way of accuracy between the two methods under discussion.

H. Birchard Taylor. The writer is impressed with the speed with which accurate determinations of quantity of water can be made by means of either of the methods described. Some seventeen years ago, when the first 10,500-hp. turbine at Shawinigan Falls was tested by traverses with pitot tubes in the penstock, the preparation for the test required two weeks. From ten days to two weeks were required to make the readings and about one month to work up the results completely.

The writer is satisfied that with reasonable lengths of conduits either the Gibson or Allen method is extremely accurate, but he is not prepared to state at this time what accuracy would be obtained with units involving short intake passages. The problem of properly testing a water wheel has now been reduced to a comparatively simple matter and the total cost incurred in a complete test under the new methods is far less than the costs involved with the older methods, not only because of the shorter time required but because the interferences with the normal operations of a plant are now reduced to a minimum.

Thomas H. Hogg.[1] The Gibson method is a method of measuring flow and is not a method of determining the efficiency of hydraulic turbines, although it is primarily devised to measure flow in connection with efficiency tests. So far as the measurement of flow is concerned, provided that a steady load is on the machine or a steady condition of flow exists at the instant of closure of valve or gate, there can be no question as to the accuracy of the results. There is, however, a question regarding the measurement of power. In tests of units of 65,000 hp. with a flow of 2300 cu. ft. per sec. at a head around 300 ft., we have found certain discrepancies which, when analyzed, were due to the measurements of power. In some cases the frequency of the system was changing, and with units of that capacity and with the tremendous wr^2 tied up in the rotor, a considerable amount of power will be absorbed or given off at each change of speed. This statement is not a criticism of the method

[1] **Assistant Hydraulic Engineer, Hydro-Electric Power Commission, Toronto, Canada. Discussion refers to Paper No. 1903.**

of measuring the water. It is simply a fact that must be recognized in determining the efficiency of hydroelectric plants. The measurement of the water is absolutely accurate, but the output of the machine must be adjusted by taking the load readings and adjusting for change of frequency.

In Par. 67 the author mentions the possibility of using, under certain conditions, two instruments at separate points on the pipe line to secure two diagrams, the area of one to be subtracted from the area of the other. In attempting this method at the Ontario Power Company, we found considerable difficulty in synchronizing the diagrams. The writer believes that the differential method may be utilized by attaching one instrument to two different points of the penstock.

In regard to the advantages of the Gibson method, it has effected a tremendous saving to the Ontario Hydro-Electric Commission. It has been used to test an aggregate turbine capacity upward of 900,000 kw. in three of the largest plants controlled by the Commission, and also in testing a large number of smaller plants ranging from 1200 to 6000 hp. The economic importance of accurate methods of measuring flow may be gaged from the fact that within three years there will be installed at Niagara nearly 1,000,000 hp. on the Canadian side. We are limited to a certain flow from the Niagara River and are required by the engineers of the control board instituted by the United States and Canadian governments to perform satisfactory tests on the individual units at Niagara to determine the flow in each at different gates and under different load conditions. The results of these tests will determine the amount of power that can be taken from the Niagara River. Now one per cent of 1,000,000 hp. is a considerable amount of power. Narrowing the limits of variation will mean the addition of considerable revenue. One of the best indexes to the value of the Gibson method is the expressed satisfaction of the government engineers as to its accuracy.

William Monroe White. Tests of hydraulic-turbine installations by accurate methods such as those described by Professor Allen and Mr. Gibson will have a decided effect upon the quality and performance of hydraulic-turbine machinery, because they will enable us to determine, at the installation, whether a certain shape of runner blade, draft tube, or cone center has a beneficial effect, and will give the manufacturers a check on the tests made on small models. This will tell them whether or not slight changes which show such radical improvements in the small shop models will have the same relative effect in the power house.

Heretofore we trusted absolutely to the electrical output as recorded by calibrated instruments. Recent tests by both the Gibson and Allen methods have proved that the water can be measured more accurately and more reliably than the electrical

output. This fact alone certifies to the great improvement that these men have made in hydraulic-testing methods.

The two methods are not antagonistic. For a particular installation, if a thorough study is made it will undoubtedly be found that one of the methods will be most readily applicable. Each has its advantages and has been found to give excellent results.

Authors' Closures

Charles M. Allen and Edwin A. Taylor. Referring to Professor Angus' question concerning the weir coefficient, an item also mentioned by Mr. Barrows, it was found in the laboratory test described in the first part of the paper that the determinations by the salt velocity method agreed with the venturi-meter readings throughout, and this meter had been calibrated by the weighing tank. At low heads on the weir there was considerable divergence between the salt velocity determination and the weir readings. This divergence increased as the head on the weir decreased. We were thus led to investigate the weir by means of the 50,000-gal. weighing tank, which enabled us to accurately determine the discharge over the weir. We then found that we had been using the wrong coefficients, and that the error in the weir under the particular conditions of the test ran as high as one per cent. In all later tests the quantity by weir was taken from the calibration curve.

In regard to the electrical results of an efficiency test, the usual run with the salt velocity method is from 5 to 10 minutes, during which time from 5 to 20 observations are made on both load and discharge. The average should give very accurate results.

Mr. Herschel mentions the necessary lapse of several days before the results of discharge measurement are known. If only the usual accuracy of the older metering methods is desired, the discharge results by the salt velocity method can be computed while the salt is going down the pipe by counting the seconds either on the chart or by a stop watch. A preliminary discharge curve is always plotted as the test is run and a final discharge curve is frequently prepared during the evening of the test.

Mr. Barrows mentions the applicability of the salt velocity method to open flumes. Since the paper was written, some commercial tests and also some extensive laboratory investigations have been made on open flumes. A flume 15 ft. wide, about 17 ft. deep, and 100 ft. long was used on some efficiency tests with very satisfactory results. The laboratory flume was 4½ ft. by 5 ft., and 28 ft. long. The results by the salt velocity method checked the Q by weir very closely. A permanent installation is now being designed for an open conduit in Massachusetts.

Mr. Johnson refers to the color velocity method of water measurement. From experience with both the color velocity and the salt velocity methods, the authors believe the latter to be the more accurate. The graphical record of the salt curve is a very desirable

feature and if some method of recording the entire color curve could be used, it would increase the accuracy and value of that method.

The authors appreciate Mr. Riddile's presentation of some of the details and some excellent sketches of conditions on a test of the 20-ft. concrete penstock at Shawinigan Falls. This method of checking results by measuring the *same* water in different sections of a conduit was recently used on a test at the New England Power Company's plant at Searsburg, Vt. There the regular efficiency runs were made on a 6-ft. 6-in. steel penstock 500 ft. long, and some check runs were made on an 8-ft. wood-stave pipe 18,000 ft. long. The results check within one-half of one per cent.

Mr. Taylor is impressed with the speed of the newer methods of measuring water as compared with the older ones. Last year one of the authors spent just two days at a plant in Vermont, made complete tests on two units — one at six gate openings and the other at four — and left the discharge results with the company's engineer.

Mr. Taylor also mentions that interference with the operating conditions while testing a plant can now be reduced to a minimum. In 1922 five salt velocity tests were completed without emptying the penstock or even shutting down the wheels.

The authors agree with Mr. White that, under present conditions, the water can be measured more accurately than the electrical output.

N. R. Gibson. Mr. Herschel has apparently overlooked the means employed when he refers to the water measurements under discussion as essentially methods of work. He refers to the venturi meter, the excellent qualities of which have not been denied. But since this question has been raised — although it is not strictly included in the subject under discussion — it should be pointed out that the venturi meter becomes impracticable in modern power plants where the conduits are large. The permanent loss of head cannot be fully annulled by the use of bypasses as suggested, because bends and the enlargement and contraction of the section of the conduit are always accompanied by loss of head. The expense of a temporary meter built of wood inside the conduit becomes prohibitive for large conduits. Furthermore, the venturi meter installed in a penstock does not record the flow correctly except under conditions of steady flow. The frequent adjustment of the turbine gates by the governor in a hydroelectric plant causes changes of pressure in the penstock which disturb the venturi-tube differential, because the change of pressure at the downstream piezometer is not of the same magnitude as the change at the upstream piezometer. This disturbance is relatively greater when the distance between the two piezometers is an appreciable part of the whole length of the conduit. The fact must not be overlooked that a water wheel is an excellent water meter when it has been calibrated by determining either the gate opening-discharge

relation or, in the case of hydroelectric power plants, the power-discharge relation.

Professor Angus has evidently assumed that the electrical measurement in a test is made simultaneously with the water measurement and, therefore, a single electrical reading may be in error if the speed of the unit should happen to be varying. As explained in Pars. 14 to 17 of the paper, the gates of the turbine are fixed during a run and power may be observed as long as desired. Usually several readings of power are taken for a period of two minutes, employing a standard indicating meter, but these may be supplemented by readings from a rotating standard integrating meter, and the duration of these readings may be made to suit any conditions. The variation in flow when the gates are fixed is inappreciable within the limits of speed changes which occur under test conditions. Frequently it is found convenient to plot the gate-opening-discharge relation and the gate-opening-power relation as shown in the example given in the discussion by Mr. Strowger, and from these two curves the power-discharge relation may be determined. Usually, however, three runs at each gate opening will give an average value which will eliminate any discrepancies in power readings.

The discussions by Mr. Kruse and Mr. Strowger are based on their personal experience in the use of the method.

Mr. Johnson has drawn an interesting comparison between the volumetric feature of Professor Allen's method and the gravimetric feature of the author's method, but his conclusion that both are much alike in principle is astonishing. It would be difficult to imagine two methods more unlike in principle, the one determining the elapsed time required for a substance in the water to pass between two fixed points, and the other determining the product of pressure and time during the interruption of the flow. In regard to the surge-tank phenomena referred to by Mr. Johnson, as he states that its function in relation to water measurement was not thought of until the author demonstrated his method, it might be noted that there is a marked difference between working from a known velocity in the conduit to the surges in the surge tank and working in the reverse order from the surges to the unknown velocity. A part of the change of level in the tank is due to the change in friction and velocity heads in the conduit, and must be accounted for in some manner such as that used by the author.

Mr. Hogg refers to the difficulty experienced in using two diagrams taken at different points on the penstock. There is considerable difficulty in synchronizing two diagrams, especially when the gradient in the main conduit is affected by surges in a surge tank. A synchronizer has been devised for the purpose, but, as pointed out by Mr. Hogg, the development of the differential diagrams shown in Fig. 11 of the author's paper has led to so many advantages that there will probably not be many occasions when two diagrams will be required.

No. 1904

FLOW OF WATER IN SHORT PIPES

By O. W. Boston,[1] Ann Arbor, Mich.
Member of the Society

The primary purpose of this paper is to present the method employed by the author in determining the frictional resistance or loss of head for water flowing in pipes. It is believed that this method, together with the mathematical determination of results, can be made use of in work of this nature at a great saving in time where tests in quantities are made for comparative analysis.

Data for mathematical interpretation are obtained by allowing a tank full of water to empty, the water flowing through the tube, pipe, valve, etc., calibrated. In this way all velocities within the head range are available in one test. The apparatus is simple and compact, making large weighing tanks and scales unnecessary.

Values for the coefficient of discharge of short tubes of diameters from 1 to 8 in., inclusive, are computed and tabulated for heads up to about seven feet.

During the academic year of 1916–1917 a series of experiments was undertaken at the University of Michigan to determine coefficients to represent the frictional resistance of the flow of water in pipes. Only the results for short tubes were completed, however, when the author entered the naval service. It is believed that the method employed in these experiments could be used to advantage if more commonly known, and because of its simplicity, brevity, and ready adaptability to many problems of this character, the mechanical and mathematical methods of investigation are presented below, along with a statement of the results obtained. Once a set-up is completed, many tests of a comparative nature may be run in a short time.

2 In order to determine accurately the discharge of any pipe, the conditions under which the pipe is discharging must be known. These conditions are so varied and complicated that theoretical formulas give only approximate results. Accurate results may be had for every particular case, however, if the several quantities involved are modified by coefficients or exponents which have been predetermined for specific cases by experimental work.

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Contributed by the Power Division and presented at the Annual Meeting, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

Nature of the Tests Undertaken

3 In the case of a straight pipe flowing full and discharging under a constant head into the atmosphere, the only losses in head to be considered are the head loss at the entrance, the head loss due to friction between the pipe and the moving water, and the head loss due to the internal friction of the water. It was intended through these experiments to determine for pipes ranging in even inches from 1 in. to 8 in. in internal diameter, inclusive, coefficients to represent

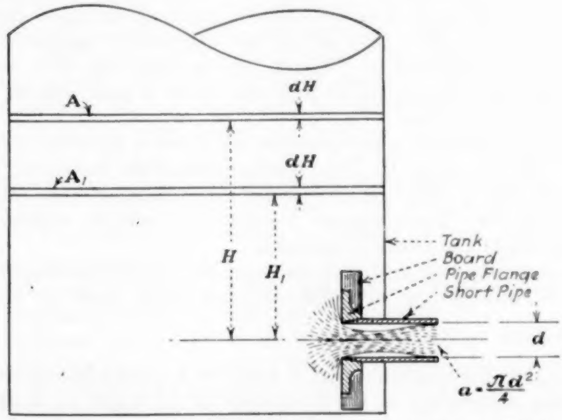

Fig. 1 Tank and Short Pipe as Arranged in Tests

a Entrance loss
b Friction loss per foot
c Friction loss due to bends of various radii
d Sudden-enlargement loss
e Sudden-contraction loss
f Friction loss of various types of valves.

With the apparatus selected, these coefficients could be obtained for all heads between 0 and 6.8 ft. (i.e., for velocities between 0 and 21 ft. per sec.).

4 A set of discharge constants for all heads within the limits was obtained for each pipe diameter in one test by allowing a filled tank to empty itself, the water flowing out through the orifice, tube, or pipe as in Fig. 1, which shows a vertical section of the tank and short pipe. For simplicity the notation used is as follows:

H = head of water above center of outlet, ft.
A = cross-sectional area of tank at H, sq. ft.
dH = drop in head in a small increment of time (dT seconds), ft.
dT = time for the head to be lowered the distance dH, sec.

a = cross-sectional area of pipe (or outlet), sq. ft.

C = coefficient of discharge, which is the relation between actual and theoretical discharge

g = acceleration due to gravity, ft. per sec. per sec.

v = average velocity in tube (ft. per sec.) for the head H, which equals the theoretical velocity modified by the constant C, or $v = C\sqrt{2gH}$; also $v_1 = C_1\sqrt{2gH_1}$, etc.

Then

$$\frac{dH}{dT} A = \text{discharge } Q \text{ at head } H \text{ in cu. ft. per sec.}$$

also

$$\frac{dH_1}{dT_1} A_1 = \text{discharge } Q_1 \text{ at head } H_1 \text{ in cu. ft. per sec.}$$

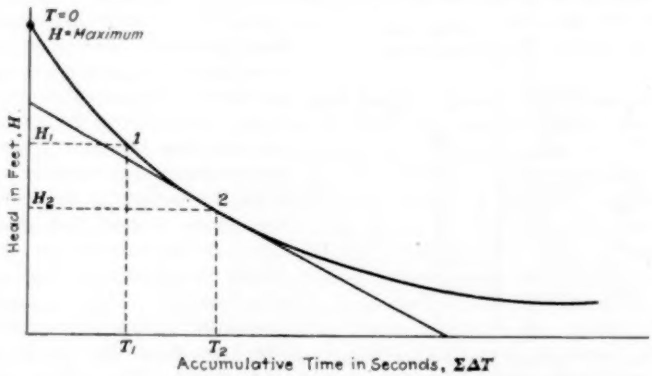

FIG. 2 HEAD-TIME CURVE OF FLOW THROUGH PIPE

5 The discharge through the short pipe for head H must also equal $Q = av = aC\sqrt{2gH}$; also $Q_1 = aC_1\sqrt{2gH_1}$ at head H_1. Therefore

$$\frac{dH}{dT} A = aC\sqrt{2gH} \quad \text{and} \quad \frac{dH_1}{dT_1} A_1 = aC_1\sqrt{2gH_1}$$

6 When the flow through the opening is started, the time equals zero and the head is a maximum. As the time is increased the head is reduced and follows a curve similar to that shown in Fig. 2. Also T_1 is the time required for the head to be reduced to H_1, T_2 the time for it to be reduced to H_2, etc. It is therefore necessary only to measure experimentally this relation between the head and time to determine a coefficient of discharge for any opening. The value of (dH/dT) at any head H is the value of the

tangent to the H-T curve at that point, and may be determined as shown in Fig. 2 or by analytical methods.

The Apparatus Used

7 The apparatus used consisted of a cylindrical cypress tank 8 ft. deep and 8 ft. in diameter, a head-time recording instrument, and a set of short pipes. After the tank was well soaked its cross-sectional area for each tenth of a foot above the center of the tube was determined.

8 The head-time recording instrument consisted of a motor-driven drum 8 in. in diameter and 20 in. long. On the drum was mounted the data sheet. Two pens rested on the data sheet in such a way that, as the drum was rotated, two parallel lines were drawn. Also, as the drum rotated the pens were moved along its axis so that the parallel lines were really helices on the drum. One of the pens was attached by an electric circuit to a seconds clock so that each half-second was indicated. The other was attached by an electric circuit to a float on the surface of the water. The float was attached to a fine wire wound on a small drum. As the float was lowered with the surface of the water, the drum rotated and closed the electric circuit for each tenth of a foot that the float was lowered. These two pens produced parallel lines with steps as shown in Fig. 3. In this figure $dT = \Delta T$ or the time for the surface of the water to be lowered $dH = \Delta H$ or 0.1 ft.

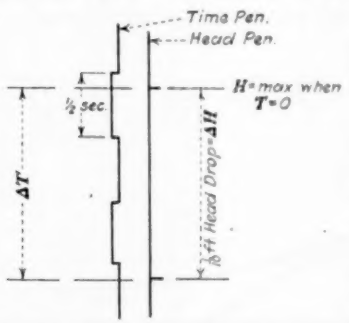

Fig. 3 Lines Drawn by Pens of Head-Time Recording Instrument

9 In running an experiment the point on the head line for initial or maximum head was noted and the time designated as zero. From this point on, the head was reduced 0.1 ft. for each contact; the lapsed time for each 0.1-ft. change in head was calculated and the individual times then accumulated. These values plotted give the head-time curve of Fig. 2.

10 The short tubes or pipes were approximately 1, 2, 3, 4, 5, 6, 7, and 8 in. in internal diameter, respectively, and their length was made three times their diameter. According to an experiment by Professor Pardoe of the University of Pennsylvania, published in *Engineering News*, September 14, 1916, the length at which a 2-in. tube flows full varies with the head. The 2-in. tube of his experiment flowed full at a length of approximately six inches, or three times the diameter, under a head of 8 ft. The experiment also showed that this length decreased as the head was lowered.

11 In general, water will enter a short pipe under the same conditions as it discharges through an orifice. Just inside the pipe the area of the stream is contracted so that the pipe does not flow full. Farther along the pipe this area is increased to that of the pipe and the pipe flows full (see Fig. 1). A partial vacuum exists in the tube where the pipe does not flow full, and it is this partial vacuum that increases the discharge of the short pipe over that of the orifice. The discharge coefficient is increased from about 0.6 for a circular sharp-edged orifice to about 0.8 for the short tube.

12 As shown in Fig. 1, the short pipe was screwed into a standard flange so that the end of the pipe was flush with the face of the flange. The face and internal diameter of the pipe were finished to present a knife-edged entrance (orifice) into the pipe. Each flange was recessed in a board which had been boiled in paraffin so that the surfaces of flange and board were flush, all irregularities on this surface having been filled with paraffin. The object in providing these boards was to present a plane surface from three to four diameters of the pipe wide on all sides of the entrance to eliminate as much as possible unnecessary eddies and to standardize conditions.

13 The board with the pipe and flange in place was screwed to the inside of the tank, near the bottom, the pipe extending horizontally through the wall of the tank as indicated in Fig. 1. The opening between the pipe and tank was stuffed to prevent leakage. While the tank was being filled with water, a piece of leather belting was held in position over the orifice by the water pressure. The tank was filled to a specific height above the center of the orifice and allowed to stand many hours to permit the water to come to rest. A wire was attached to the belting over the orifice so that it could be removed from above when it was desired to start the test without causing excessive motion of the water in the tank. This method allowed the water, while being discharged, to settle slowly in the tank with very little disturbance, and because of the large ratio between the orifice and tank areas, the results were but little affected by the velocity of approach.

The Tests

14 The 1-in. short pipe was used in the initial set-up. The first time that the water was emptied through it the pipe did not flow full but acted as an orifice, the stream of water having the appearance of a solid glass rod. The discharge was then stopped, by plugging the tube on the discharge end, and subsequently released. This caused the pipe to flow full at the discharge end as was shown by the data, which gave the values of an orifice in the first case and of a short pipe in the second. The fact that the 6-in. and 7-in. short pipes discharged as pipes in some cases and as orifices in others, while the 8-in. pipe discharged only as an orifice, indicates that as the diameter of the pipe is increased the ratio

TABLE 1 COEFFICIENTS OF DISCHARGE FOR SHORT PIPE 1 IN. IN DIAMETER

$$C = \frac{\frac{dH}{dt}}{a\sqrt{2gH}}$$

Diameter of pipe, 1.029 in.
Length of pipe, 3.000 in.
Area (a) of pipe, 0.005775 sq. ft.
Temperature of water, 58 deg. fahr.
Gravity, 32.16 ft. per sec. per sec.

H	A	A_a	ΔT	Δt	$\Sigma\Delta t$	$\frac{dH}{dt}$	C
6.8	39.34					−0.0957	0.792
6.7	39.4	39.37	48.6	1.235	1.235	−0.0949	0.792
6.6	39.46	39.43	49.2	1.234	2.478	−0.09415	0.791
6.5	39.52	39.49	38.2	0.968	3.446	−0.0933	0.791
6.4	39.585	39.552	46.2	1.169	4.615	−0.0927	0.791
6.3	39.647	39.61	41.4	1.046	5.661	−0.0920	0.791
6.2	39.71	39.673	40.8	1.030	6.691	−0.0911	0.791
6.1	39.77	39.74	44.1	1.111	7.802	−0.0904	0.791
6.0	39.835	39.802	46.2	1.160	8.962	−0.0897	0.791
5.9	39.90	39.87	45.0	1.129	10.091	−0.0888	0.791
5.8	39.96	39.93	44.4	1.112	11.203	−0.0880	0.791
5.7	40.02	39.99	44.4	1.111	12.314	−0.0873	0.791
5.6	40.10	40.06	48.0	1.200	13.514	−0.0866	0.791
5.5	40.17	40.135	45.85	1.144	14.658	−0.0859	0.791
5.4	40.245	40.21	47.1	1.173	15.831	−0.08515	0.791
5.3	40.32	40.29	45.8	1.138	16.969	−0.0843	0.791
5.2	40.39	40.36	50.3	1.247	18.216	−0.0835	0.791
5.1	40.464	40.425	46.8	1.160	19.376	−0.0827	0.791
5.0	40.54	40.50	50.4	1.245	20.621	−0.0819	0.791
4.9	40.61	40.575	48.6	1.199	21.820	−0.081	0.791
4.8	40.68	40.645	51.5	1.268	23.088	−0.0802	0.791
4.7	40.76	40.72	48.0	1.180	24.268	−0.0793	0.791
4.6	40.83	40.795	53.4	1.308	25.576	−0.0785	0.791
4.5	40.90	40.865	52.75	1.290	26.866	−0.0777	0.791
4.4	40.98	40.94	54.0	1.320	28.186	−0.0767	0.791
4.3	41.05	41.015	51.6	1.260	29.446	−0.0758	0.791
4.2	41.13	41.09	55.65	1.356	30.802	−0.075	0.791
4.1	41.20	41.165	55.2	1.375	32.177	−0.07441	0.791
4.0	41.28	41.24	56.25	1.364	33.541	−0.0732	0.791
3.9	41.35	41.32	54.6	1.322	34.863	−0.0723	0.791
3.8	41.42	41.385	57.0	1.379	36.242	−0.0714	0.791
3.7	41.50	41.46	59.4	1.434	37.676	−0.0704	0.791
3.6	41.58	41.54	61.7	1.485	39.161	−0.0695	0.791
3.5	41.66	41.62	58.2	1.400	40.561	−0.0685	0.791
3.4	41.74	41.70	59.4	1.426	41.987	−0.0696	0.791
3.3	41.82	41.78	63.0	1.508	43.495	−0.0666	0.791
3.2	41.90	41.86	69.6	1.660	45.155	−0.0656	0.791
3.1	41.98	41.94	60.0	1.431	46.596	−0.0646	0.791
3.0	42.06	42.02	65	1.547	48.133	−0.0634	0.790
2.9	42.14	42.10	66	1.570	49.703	−0.0625	0.790
2.8	42.22	42.18	72	1.710	51.413	−0.0614	0.790
2.7	42.30	42.26	68.4	1.620	53.032	−0.0603	0.790
2.6	42.39	42.345	66.6	1.573	54.606	−0.0592	0.790
2.5	42.48	42.435	71.7	1.690	56.296	−0.0581	0.790
2.4	42.57	42.525	75.6	1.780	58.076	−0.0568	0.790
2.3	42.66	42.615	73.8	1.733	59.809	−0.0556	0.790
2.2	42.75	42.705	76.8	1.800	61.609	−0.0545	0.790
2.1	42.84	42.795	80.4	1.879	63.488	−0.0532	0.790
2.0	42.93	42.895	79.85	1.863	65.351	−0.0518	0.790
1.9	43.02	42.975	84	1.955	67.306	−0.0505	0.790
1.8	43.10	43.06	88.75	2.061	69.368	−0.0491	0.790
1.7	43.20	43.15	89.3	2.072	71.439	−0.0478	0.790
1.6	43.29	43.245	90.25	2.090	73.520	−0.0462	0.790
1.5	43.38	43.335	93	2.150	75.679	−0.0448	0.790
1.4	43.47	43.425	99.3	2.290	77.969	−0.0432	0.789
1.3	43.56	43.515	98.1	2.255	80.224	−0.0417	0.788
1.2	43.65	43.605	110	2.528	82.752	−0.0400	0.788
1.1	43.74	43.695	109.2	2.500	85.252	−0.0382	0.788
1.0	43.83	43.785	119.3	2.723	87.975	−0.0365	0.788
0.9	43.92	43.815	126.6	2.884	90.859	−0.0346	0.787
0.8	44.01	43.965	126.8	2.883	93.742	−0.0326	0.787
0.7	44.10	44.055	134.7	3.060	96.802	−0.0304	0.787
0.6	44.20	44.15	153	3.470	100.272	−0.0282	0.786
0.5	44.18	44.19	172.8	3.910	104.182	−0.0257	0.785
0.4	44.28	44.23	196.2	4.440	108.622	−0.0229	0.784
0.3	44.38	44.33	186.6	4.215	112.837	−0.0198	0.782
0.2	44.48	44.43	241.2	5.430	118.267	−0.0161	0.778

between length and diameter should also be increased. This, however, is in itself a subject for research. It is believed that the friction derived from the error in using a ratio of 3 to 1 between length and diameter instead of the theoretically correct ratio has not appreciably affected the results obtained. The 8-in. pipe when discharging as an orifice gave results which check very well the values given by other experimenters.

Computation of Discharge Coefficients

15 The method used in computing coefficients of discharge from the time-head data obtained in the tests is shown in Table 1 for the 1-in. short pipe, where

TABLE 2 METHOD USED IN COMPUTING dH/dt FOR THE 1-IN. SHORT PIPE

H	$\Sigma\Delta t$	ΔH	Δt	$\frac{\Delta H}{\Delta t}$
6.8	0			
		−0.3	3.446	−0.087057
6.5	3.446			
		−0.5	5.516	−0.090645
6.0	8.962			
		−0.5	5.696	−0.087781
5.5	14.658			
		−0.5	5.963	−0.083850
5.0	20.621			
		−0.5	6.245	−0.080064
4.5	26.866			
		−0.5	6.675	−0.074906
4.0	33.541			
		−0.5	7.020	−0.071225
3.5	40.561			
		−0.5	7.572	−0.066033
3.0	48.133			
		−0.5	8.163	−0.061252
2.5	56.296			
		−0.5	9.055	−0.055218
2.0	65.351			
		−0.5	10.328	−0.048412
1.5	75.679			
		−0.5	12.296	−0.040664
1.0	87.975			
		−0.5	16.207	−0.030851
0.5	104.182			
		−0.3	14.085	−0.021299
0.2	118.267			

H = head, ft.
A = actual cross-sectional area of tank at H, sq. ft.
A_a = average cross-sectional area of tank at H as the water is lowered 0.1 ft., sq. ft.
ΔT = time in seconds for water to be lowered 0.1 ft. at H and through area A_a
Δt = time in seconds for water to be lowered 0.1 ft. at H, based on 1 sq. ft. cross-sectional area of tank

$$= \frac{\Delta T}{A_a}.$$

$\Sigma\Delta t$ = accumulated sum of Δt at H

$$\frac{dH}{dt} = \text{tangent of } H\text{-}\Sigma\Delta t \text{ curve at } H$$

C = coefficient of discharge of pipe at H.

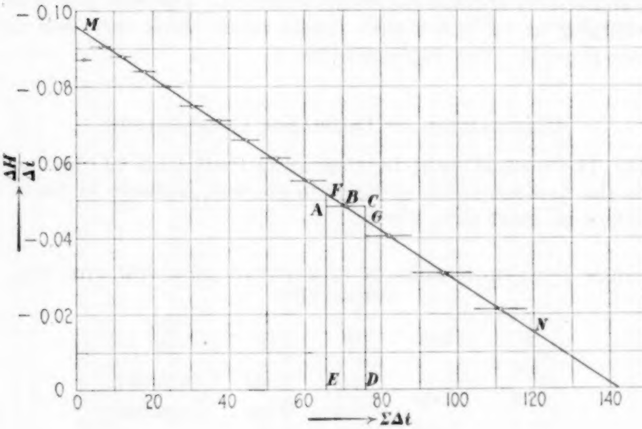

FIG. 4

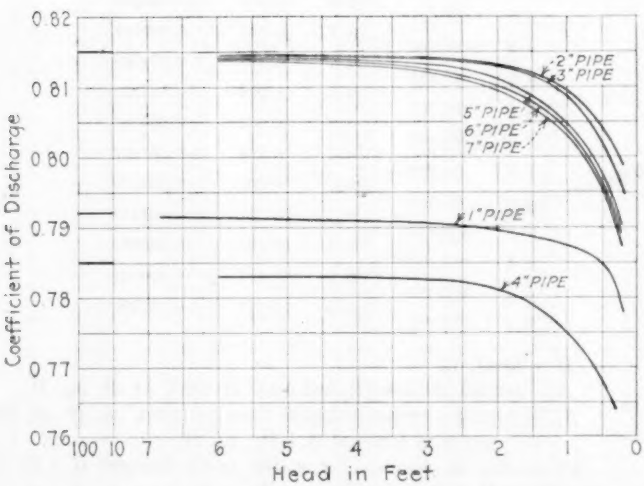

FIG. 5 CURVES OF DISCHARGE COEFFICIENTS OF SHORT PIPES

16 The coefficients given for all pipes are based on values of dH/dt as determined mathematically. Table 2 indicates the necessary steps. Values of the head are taken at random, in this case every half-foot when convenient. The total lapsed time $\Sigma\Delta t$ at that head is placed in the second column. The third

and fourth columns contain the increments of the head, ΔH, and the time, Δt, respectively. The next column lists $\Delta H/\Delta t$, which is the average slope between the points. These values are then plotted as ordinates with $\Sigma\Delta t$ as abscissas, as shown in Fig. 4. As the line AC represents the average slope of the H-$\Sigma\Delta t$ curve between points E and D, then some line FG will represent the actual slope at all points of the H-$\Sigma\Delta t$ curve within these limits, as the area $ACDE$ must equal the area $FGDE$. In other words, area AFB must equal area BCG. FG happens to be a straight line. Therefore large increments ($\frac{1}{2}$ ft.) may be taken to get the slope curve. The derivative of the H-T curve is dH/dt or the slope which equals the ordinate in Fig. 4. Also the area of all such

TABLE 3 VALUES OF DISCHARGE COEFFICIENTS FOR SHORT PIPES RANGING IN DIAMETER FROM 1 IN. TO 7 IN., INCLUSIVE

(NOTE: The 8-in. pipe functioned as an orifice.)

Head in feet	Diameter of Pipe in Inches (Length = 3 × Diameter)							
H	1.029	2.0459	3.2236	4.0556	5.0624	6.0576	6.98	7.955
100	0.792	0.815	0.815	0.785	0.815	0.815	0.815	0.601
10	0.792	0.814	0.814	0.785	0.814	0.814	0.814	0.598
6.8	0.792							
6.6	0.791							
6.4	0.791							
6.2	0.791							
6.0	0.791	0.814	0.814	0.783	0.814	0.814	0.814	0.598
5.8	0.791	0.814	0.814	0.783	0.814	0.814	0.814	0.598
5.6	0.791	0.814	0.814	0.783	0.814	0.814	0.814	0.598
5.4	0.791	0.814	0.814	0.783	0.814	0.814	0.814	0.598
5.2	0.791	0.814	0.814	0.783	0.814	0.814	0.813	0.598
5.0	0.791	0.814	0.814	0.783	0.814	0.814	0.813	0.598
4.8	0.791	0.814	0.814	0.783	0.814	0.814	0.813	0.598
4.6	0.791	0.814	0.814	0.783	0.814	0.814	0.813	0.598
4.4	0.791	0.814	0.814	0.783	0.814	0.814	0.813	0.598
4.2	0.791	0.814	0.814	0.783	0.814	0.813	0.813	0.598
4.0	0.791	0.814	0.814	0.783	0.814	0.813	0.813	0.598
3.8	0.791	0.814	0.814	0.783	0.814	0.813	0.813	0.598
3.6	0.791	0.814	0.814	0.783	0.814	0.813	0.813	0.598
3.4	0.791	0.814	0.814	0.783	0.814	0.813	0.813	0.598
3.2	0.791	0.814	0.814	0.783	0.813	0.813	0.812	0.597
3.0	0.790	0.814	0.814	0.783	0.813	0.813	0.812	0.597
2.8	0.790	0.814	0.814	0.782	0.813	0.812	0.812	0.597
2.6	0.790	0.814	0.814	0.782	0.813	0.812	0.811	0.596
2.4	0.790	0.814	0.813	0.782	0.812	0.812	0.811	0.596
2.2	0.790	0.813	0.813	0.782	0.812	0.811	0.810	0.595
2.0	0.789	0.813	0.813	0.781	0.811	0.810	0.810	0.595
1.8	0.789	0.813	0.813	0.780	0.810	0.809	0.809	0.594
1.6	0.789	0.812	0.812	0.780	0.809	0.808	0.808	0.594
1.4	0.789	0.812	0.811	0.778	0.808	0.807	0.806	0.593
1.2	0.788	0.811	0.810	0.777	0.807	0.806	0.805	0.592
1.0	0.788	0.810	0.809	0.775	0.805	0.804	0.803	0.590
0.8	0.787	0.808	0.807	0.773	0.803	0.802	0.801	0.587
0.6	0.786	0.806	0.804	0.770	0.800	0.798	0.798	0.582
0.4	0.784	0.803	0.800	0.766	0.796	0.794	0.793	
0.2	0.778	0.800	0.796					

rectangles as $ACDE$ must equal the area under the line MN. In this way the line MN is located and its equation found to be

$$\frac{dH}{dt} = 0.000672\,t - 0.0957 \quad \text{[1]}$$

Let $t = \Sigma\Delta t$. Integrating this curve and making it pass through some point on the H-$\Sigma\Delta t$ curve, gives

$$H = 0.000336t^2 - 0.0957t + 6.822 \quad \text{[2]}$$

Solving with Equation [1],

$$\left(\frac{dH}{dt}\right)^2 = 0.001344\ H - 0.0000103 \quad . \quad . \quad . \quad . \quad . \quad [3]$$

which gives a relation between the tangent dH/dt and the head H, from which the values of dH/dt as given in Table 1 were computed.

Results

17 Table 3 gives the coefficients of discharge for all of the short pipes. The values for the 8-in. pipe are those of an orifice, indicating that the pipe did not flow full. These values are plotted in Fig. 5. The values of the 1-in. and 4-in. pipes are consistently lower than those of the other sizes, for which no explanation can be given. The curves are similar, which indicates a constant error, such as might occur in the measurement of the pipe area. The experimental material had been removed in the author's absence of four years, so that measurements could not be checked or experiments continued.

18 It is interesting to note that the coefficients do not decrease when the head is greater than 3 ft. to 5 ft., as is the case with coefficients of discharge of orifices. Fig. 5 shows the values based on formulas for heads of 10 ft. and 100 ft.

19 The author wishes to express his appreciation to Mr. H. W. King, professor of hydraulics, in whose department the experiments were conducted, and to Dr. T. R. Running, professor of mathematics, for his inspiring suggestions and interest in the experiments and the mathematical interpretation of the data.

DISCUSSION

R. W. Angus. The purpose of the work described is not quite clear, as the general values of the coefficients for short tubes have long been known, as well as the fact that the coefficient of discharge varies with the head and with the size of the pipe. The paper does not state the size of the disk on the end of the pipe, nor the height of the tube above the bottom of the tank, so that it is not clear whether contraction is complete or not.

The experiment is one where the flow is unsteady, and there are variations in the head and in the coefficient of discharge as recorded in the paper, but it is possible that there is a time lag between the coefficient and the head at which it was obtained, and no evidence is presented which proves that the coefficient for steady flow under a given head would be the same as the one found by such an experiment as the paper describes.

If the coefficients can be shown to be the same as would be obtained for steady flow under the same heads, the method would be valuable in rating large weirs in cases where there is no available reservoir large enough to make volumetric measurements. If a volume of water could be run from a tank over a weir and reliable

coefficients determined for each head in the way described in the paper, then a relatively small tank of water could be used to rate a large weir.

L. F. Moody. There seems to be nothing in the paper to indicate any analysis of the pressure at the vena contracta in the pipe. In measuring the discharge through a short tube, one of the important variables involved is the pressure at this point. As the head on the orifice is increased that pressure will decrease more and more below atmospheric in proportion to the velocity head created, and at the time when the barometric limit is approached a mere overall coefficient is not enough to predict the flow through a standard tube.

Some figures are given in the paper where the values are extrapolated to apply to a 100-ft. head. This is merely a hypothetical condition, because before that head is reached, the whole mode of flow will have changed.

The Author. Replying to Mr. Angus, the short-tube coefficients were obtained simply as the forerunner to the determination of the coefficients for the flow of water in pipes, valves, bends, etc. The paper is confined to the short tubes or pipes because the work was started just before the beginning of the war. When it was possible to resume it, the apparatus had been removed and the only data sheets available were one for each tube, so that the paper is based on a single test for each of the eight pipes. Therefore we will not vouch too much for the accuracy. However, for the 8-in. pipe which flowed as an orifice rather than as a short tube, both, according to the appearance of the water and according to the coefficients that were determined, gave results that conformed to the third decimal point with other tests for an orifice. The value of the coefficient above the 6-ft. head is admittedly questionable. All the coefficients determined on the basis of weighing the water for each test performed at a certain definite head begin to fall off after that point, whereas the results determined from the equations developed continue to rise to a certain maximum which is slightly above 0.004 for a head of 100 ft.

In regard to the disk, i.e., the flange, on the short pipe, this was increased over the ordinary diameter of the flange by a paraffin-soaked board—to increase the flat surface passed over by the water entering the pipe—to three to four times the diameter of the pipe. The object was to standardize conditions.

Replying to Mr. Moody, no attempt was made to measure the partial vacuum at the vena contracta. Other experimenters have done that work with more or less success. We anticipated difficulty with just that point in connecting long pipes to the short tubes, because there is a back-up due to frictional resistance offered in a straight pipe which probably would affect the partial vacuum in the short tube. What this would amount to would have to be determined by further experiment.

No. 1905

A STEAM-LOSS PREVENTION PLAN OPERATING IN A TEXTILE-FINISHING PLANT

By Henry Morgan Burke,[1] North Dighton, Mass.
Member of the Society

The author gives details of simple and elastic methods installed in the power department of a large textile-finishing concern, which made it possible to diagnose daily plant operation, introduce new operating practices, enthuse the personnel with the importance of their tasks, keep accurate records, and effect important economies in the generation, distribution, and utilization of steam and power.

THE importance of the elimination of waste in a power department is probably generally recognized, but few, it is believed, realize that the cost of this department is second only to the payroll in a textile-finishing plant.

2 The rising price of fuel and labor, Southern competition, and usually the inadvisability of heavy investment in more efficient equipment in the so-called "non-producing" department accentuate the importance of careful study of the generation, distribution, and utilization of steam and power.

3 For five or six years the author and his associates had studied their power plant — that of the Mount Hope Finishing Company — that was rapidly depreciating and becoming daily less suitable to meet the demands made upon it by an ever-increasing growth of the "productive plant." They stemmed the overwhelming tide from time to time with expedient methods such as the introduction of the use of fuel oil, etc., but were finally forced to face their problem squarely and solve it in the most economical manner.

4 With the stimulation of a new boiler plant, work was begun on the devising of new methods by which the daily plant operation could be diagnosed, new operating practices introduced, the personnel enthused with the importance of their tasks, accurate

[1] Plant Engineer, Mount Hope Finishing Company.

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records kept, and last but not least, schemes contrived to maintain higher efficiency without interruption.

Organization of the Power Department

5 As greater interest was shown in the power plant, it became more apparent that it was necessary to press on much farther in order to secure the maximum efficiency.

6 Operating methods previous to the installation of the new methods were not different from those in other plants of the same class. The chief engineer was responsible for uninterrupted and economical production of power, but left in every detail to his own resourcefulness.

7 Instrument equipment was limited to feedwater meters, oil meters, and draft gages, but no operating standards were established and no means available for judging how the load should be distributed among the boilers. The cost keeping employed was of the simplest nature and the only record kept was the curve of apparent evaporation by weeks, based on total water fed through the venturi meter; however, no correction was made for that part of the water not turned into steam but used up for other purposes such as blowdown, washout, and leaks.

8 With the provision of boilers with steam flow meters, oil and steam pressure gages, flue-gas analyzers, and pyrometers it was possible to run "standardization tests," and by arousing interest in the firemen and furnishing the chief engineer with the correct basis to work on, the progress made was rapid and certain, responsibilities were properly allocated, and detailed records were kept and *examined* daily. In this manner it became possible to establish an economical operating practice and standardize most of the influential conditions as to:

- *a* Distribution of load among boilers
- *b* Putting on and taking off load from boilers
- *c* Regulating draft according to fires and fires according to load
- *d* Cleaning of heating surface by removal of soot and scale
- *e* Method of firing coal-stoked boilers.

9 Inasmuch as 73 per cent of the steam generated is used directly for processing the goods and the remaining 27 per cent is used for process, heating, and warming up large quantities of water for various purposes after it is exhausted from the engines at 7 lb. pressure, the distribution of steam presented a large problem as to:

- *a* Metering
- *b* Despatching
- *c* Elimination of losses.

10 The information gained from these studies brought the management to the realization that the organization of the power department should be changed. The chief engineer was thereupon

made responsible for the generation of steam and power, and the power despatcher responsible for the despatching of the steam to the consumers and also for the elimination of losses.

The Power Despatcher

11 *Duties.* The power despatcher's job is a large one and a very satisfactory one to perform. It is a position that requires tact, but once the right man is found, tangible savings begin. His specific duties are as follows:

a Inspection of all distributing lines, and conductors (except sprinkler system), and all valves, traps, appurtenances, and instruments thereon, for the purpose of preventing any losses of heat, light, and power in transmission.

b Investigations and studies of the various manners in which steam, water, and power are used for various purposes, with the aim of devising and recommending a more economical practice.

c Recording of steam, power, and water consumption by various departments and preparation of reports and cost charges.

Studies Made by the Power Despatcher

12 An inspection revealed many and varied losses which were later turned into savings. A simple example of economies that were effected is the caring for steam traps. Where there are large numbers of these it is often wise to have one man look after traps alone, especially in the cold months of the year. If there are not a sufficient number to require the full time of a man, it has been found a good policy to have him devote a definite part of each day in attending to this equipment, and the remainder of his time to other jobs.

13 A plan of the piping system not only showed where savings could be made in the old installation but was a great help in planning how to avoid losses in future additions to the system. This knowledge at hand immediately prevented clumsy and inefficient piping layouts that might have been made had lines been put in as needed with as little regard to the existing and future possibilities as is ordinarily the case.

14 Time and space do not permit of too lengthy an exposition of piping detail. Investigation into the uses made of steam revealed the fact that the conditions existing simply abounded in opportunities for savings, and those engaged in it began to wonder why it was that so many find fault with the cost of generation when there is such waste in utilization. The most natural thing in the world, therefore, was to fix the responsibility on the consumers of steam. "If the consumer must waste steam, make him pay

for it," was the slogan adopted. Of course this was easier said than done.

Metering Power

15 Keeping in mind that too much red tape, too many records, too many charts to change daily, not to mention too much interference with the work of a department, might make the cost of the despatcher's work too great and possibly offset the savings, the work was approached on a modest scale, taking the heavy users of steam first and installing a few meters on trial in their departments. The results were so gratifying that it was deemed advisable to continue the installation of meters until every pound of steam generated could be accounted for.

16 The type of meter to be installed presented a problem since, if it became necessary to locate meters at the point where the

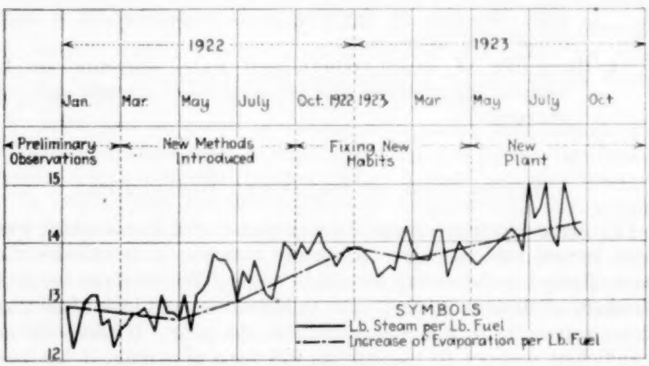

Fig. 1 Chart Showing Progress Made in Building up Efficiency of Steam Generation

steam was used and must be read daily, meter readers would be an added expense to the despatcher's department. Consequently it was decided to use an electrically operated instrument that could be centrally located in the despatcher's room. This effected not only a great saving in time and money, but permitted one man to see at a single glance how the different users were drawing on the steam supply. An unexpected draft on the boilers was instantly known and investigated by telephone. If the draft was to be permanent, orders were sent to the chief engineer that more steam would be required for a definite period. If, on the other hand, a big steam user closed down for any reason and did not advise the despatcher (and they never will if steam waste does not affect their pay), the latter immediately noticed the fluctuation, found out why, and advised the chief engineer that less steam would be required for a definite or indefinite period. This

method enables the chief engineer to effect economies that formerly were considered beyond attaining, and for this reason, if for nothing more, except possibly the accuracy of a meter, it is worth all that it costs to have a centrally located tell-tale method.

17 Of course the recording chart is a record that can be and is used to advise different departments where, when, and why steam can be saved, because, in using the recording chart in conjunction with the production figures, it can be shown why the department has consumed more steam than the standard allowable amount for a certain period. Where a bonus is paid on steam consumption such a record is a valuable assistance to the despatcher, foreman, and steam fitter, and consequently in the reduction of operating and maintenance costs.

18 The use of electrically operated meters makes it possible to record performance in different places at the same time. For instance, suppose that it is decided for the best interests of economy to install in a department or office a recording instrument in addition to the one in the despatcher's office. All that is necessary is to run an additional wire to the particular place and set up another recording meter. In some places it has been found that instead of setting a recording, integrating, and indicating panel in the despatcher's office, better results are secured by installing recording instruments in the department using the steam being metered and an additional recorder with an integrator in the despatcher's office. It is well to have an integrator and indicator in the despatcher's department as a check on the recorder, although not necessary.

Results

19 Previous to the beginning of this work the average apparent evaporation secured at the plant was 12.55 lb. of steam per lb. of fuel oil, corresponding to 72 per cent thermal efficiency. The latest record at hand indicates an efficiency of 83 per cent, or over 15 per cent improvement.

20 At the beginning of the work the estimate of possible annual savings was:

In generation	$40,000
In distribution	15,000
In utilization	25,000

At the time of writing this paper the evidence on record shows the following yearly savings:

In generation	$39,760
In distribution	12,320
In utilization	24,800

The work is still in progress. The savings enumerated above, it may be said, represent only a few items.

21 Actual progress in building up the efficiency of steam

generation, as illustrated in Fig. 1, is for convenience of study subdivided into four periods corresponding to the stages in progress: namely,

	Ratio of apparent evaporation
1 *Preliminary Observations*	
From January to March, 1922	12.55
2 *Introduction of New Methods*	
From April to September, 1922. Gradual improvement, average	13.25
3 *Fixing New Habits*	
From October, 1922, to May, 1923	13.87
4 *Extension of Plant*	
From June to August, 1923	14.21

22 During the last three months when the new extension of the company's boiler house was in service the records of monthly performance on a 24-hour basis, 7 days in the week, were as follows:

	Average evaporation, lb.	Equivalent evaporation, lb.	Thermal, efficiency per cent
June	13.9	15.01	80.1
July	14.6	15.78	83.0
August	14.4	15.49	82.5

23 However, the results obtained by research and subsequent changes made are sometimes quite difficult to calculate in dollars and cents. For example, a simple change in shutting off main pipe lines to distant parts of the plant over week ends and heating warehouses and rooms where work could be more efficiently carried on in lower temperatures, made it possible to effect an annual saving of about $3000.

24 Reduced steam pressure in caustic evaporators produced the following economies:

Average weekly caustic recovery, lb.	72,000	71,800	86,786
Steam consumption, lb.	838,462	650,109	799,515
Pounds of steam per pound of caustic	11.6	9.06	9.22
Saving, per cent		19	
Actual saving annually			$3800

25 Utilization of steam in various processes presents the most complicated yet the most promising line of research. With a steam consumption of about 710,000,000 lb. per year, the company spares no effort to work out methods to secure the same or a better grade of work with less steam. This means of course that each unit has to be studied separately. In this respect it makes no difference whether a plant is large or small, since the equipment used in large establishments is similar to that employed in very small bleacheries or laundries.

26 The results of the company's efforts in this direction may well be illustrated by examples.

a A large amount of water required in a silk dye department

was provided by two large storage heaters using live steam. Experiments established the fact that if exhaust steam were used instead of high-pressure the water could be heated to from 110 deg. fahr. to 210 deg. fahr. The following tabulation shows clearly the advantage in favor of low-pressure steam.

Steam used per week, lb.	2,870,441	2,048,035
Cost of steam per week	$1200	$810
Weekly saving		$390
Annual saving		$20,000

b On dolly washers the use of high-pressure steam to bring cold water to the boiling point was first changed by using steam of lower pressure with a resulting reduction of steam consumption of 30 per cent or a saving of $3200 annually. By supplying these washers later with hot water, preheated by means of excess of exhaust steam at a slight expense for piping, the following saving was made: Instead of heating water from 80 deg. to 210 deg. fahr. in the washer by live steam at 40 lb. pressure at the expense of 130 B.t.u. per lb., it was preheated with exhaust steam to 150 deg. fahr. and again heated up to 210 deg. at the expense of 50 B.t.u., securing 54 per cent economy or the equivalent of an annual saving of $5400.

27 It is not the prime object of the company's investigations into the uses of steam to effect changes in actual process work. Nevertheless from time to time enlightening information is brought out, such as that regarding the idle time that increases the expense burden by wasting steam and power to keep a machine in readiness during the idle period. On account of the importance of this phase of the work the company is undertaking to show idle time by means of Gantt straight-line charts.

Records

28 Records are of the utmost importance to the plant engineer to carry on the work as planned, control the progress, and maintain the interest of the men responsible for performance.

29 *General Records.* An endeavor has been made to simplify and keep to a minimum the number of forms to be made out daily and to crystallize the generation-of-steam records into one form known as the Fuel-Utilization Chart.

30 This chart, which is reproduced in Fig. 2, shows the daily performance of each shift of firemen by indicating with a straight line the evaporation of water per pound of fuel. Each day is spaced into fifteen sections, the ratio of equivalent evaporation of 15 lb. of water for each pound of fuel oil. The evaporation attained by each shift is laid off by a straight line, the length of which shows how much the shift fell short of or exceeded the standard.

31 The results for the three shifts are in like manner laid out

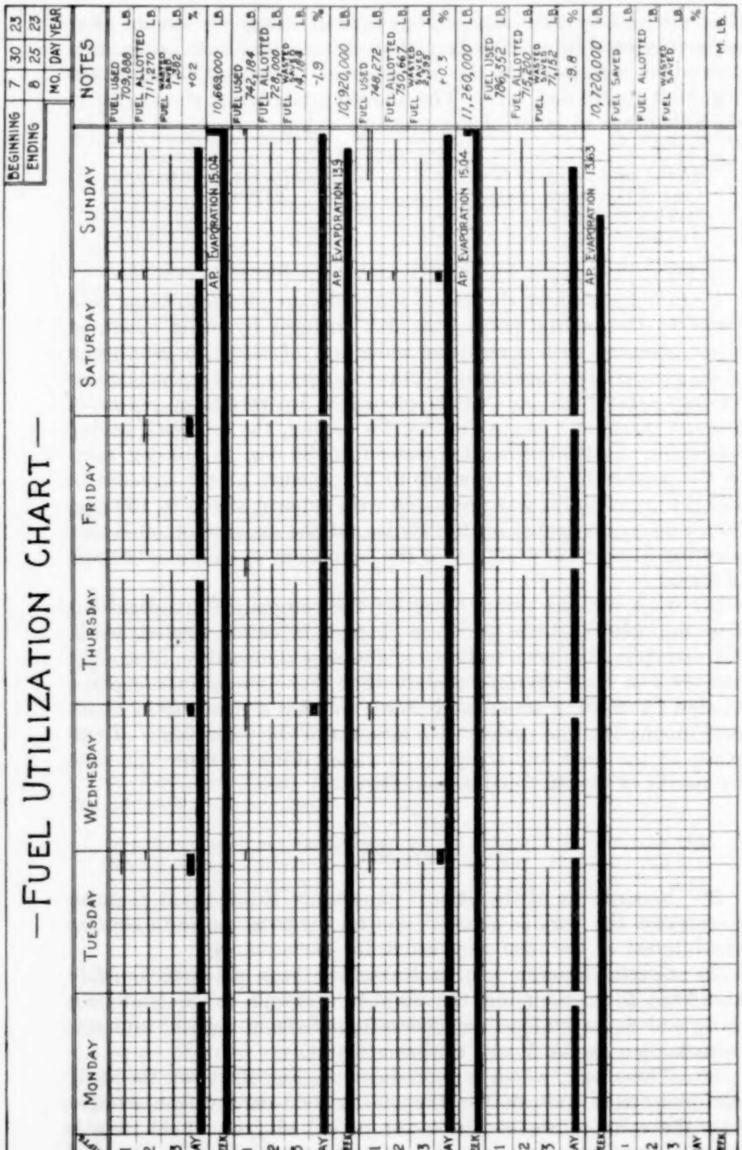

FIG. 2 FUEL-UTILIZATION CHART

with a heavier line, the length of which is the average daily evaporation. The average weekly consumption is in turn plotted, and represents the sum of the average daily results in a continuous line which shows longer or shorter according to whether the evaporation was below or above the standard set.

32 On the right-hand side of this form it will be seen that the weekly consumption of oil is checked against the weekly allowance, and the saving or loss of fuel shown. This chart is posted in the boiler room for the firemen's inspection, and it is watched with as much eagerness as if it were a baseball score.

33 *Distribution Records.* From the autographic records of the flow meters of the despatcher's office the power despatcher recapitulates, condenses, and compiles the statistics for the plant engineer on the several forms briefly described below.

34 The Power Despatcher's Summary, records the total steam generated and distributed, showing by departments the consumption, the cost per 1000 lb., and the cost per unit of merchandise processed.

35 In addition to the above a weekly Steam-Cost Record is kept for each of the departments so that their respective fluctuations can be seen at one glance. The data for this sheet are taken from the Power Despatcher's Summary.

36 One of the unique features of the system as operated is that the power plant is virtually an independent business carried on within the walls of the plant. It buys its raw product — fuel — and sells the finished product — steam — to its customers, the various departments of the plant. It pays rent for its housing and taxes, and it also has to pay for insurance and other fixed charges, labor and supplies, so consequently must charge for its finished product, steam.

37 It sends out bills weekly to each department for steam used, as shown by the steam flow meters, at cost of production. Exhaust steam costing less than high-pressure, its use is encouraged whereever practicable.

38 For the benefit of the accounting and sales department the power despatcher prepares monthly a report showing the distribution of the power-plant expense to the various departments. This is accompanied by a statement of costs of various items that make up the total monthly charge, as compared with the costs of the same items for the preceding month.

39 It is interesting to compare each month the actual cost to operate the power plant with the figures given in the chart of predetermined costs, Fig. 3, and it is remarkable how closely the actual and predicted costs coincide. This chart is made up of four curves, showing

a Total costs, including fixed charges
b Unit costs, including fixed charges
c Total operating costs, and
d Unit operating costs.

The data from which these curves were plotted were collected over a period of five years. The chart is useful in detecting errors in accounting and in allocating charges to the power department.

Conclusion

40 The intrinsic value of this system of conducting a power plant lies in its simplicity and elasticity. While it is being employed in a large steam plant costing one third of a million dollars annually to operate and furnishing steam and power to a plant covering

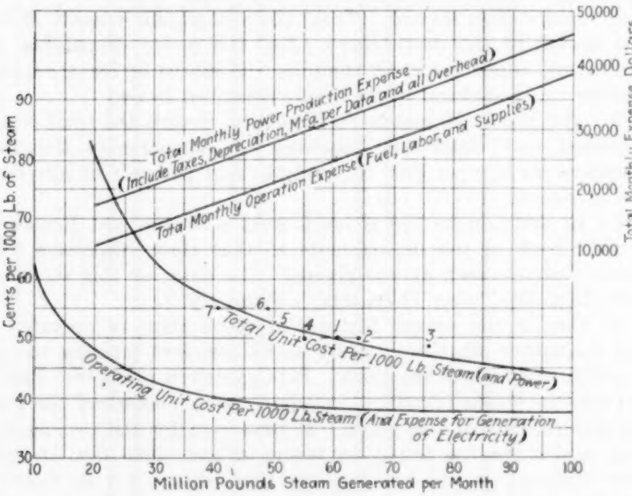

Fig. 3 Chart Showing Standard Costs of Power, Steam, and Electricity

about thirty acres of floor space, it is equally as practical for use in the small plant.

41 Such a system is surely worth while, for once the spirit of efficiency and coöperation is imbued into an organization, suggestion on the part of the management and the spirit of contest among the employees will bring about the desired results automatically.

42 These loss-prevention methods were installed in the plant of the Mount Hope Finishing Company under adverse conditions owing to the erection of a new boiler house, yet operation was not interrupted and efficiency was increased. An outstanding feature was that but one addition was made to the power-plant payroll, namely, the salary of the power despatcher, who was transferred from the engineering department office and has successfully carried on single-handed all the work of his new department.

43 In view of the fact that the plant was operated at an unusually high efficiency before the new methods were introduced, their value is clearly shown by the additional savings made.

DISCUSSION

Willard H. McGregor. The author, in a measure, precludes criticism by stating in Par. 20 that the results so far secured are composed of a few items, and that the process is still going on. The paper thus points out a method rather than a summary of concrete results.

The author gives a total use of steam of 710,000,000 lb. per year, which, with an assumed evaporation of 9 lb. per pound of coal, means 40,000 tons of 2000 lb., amounting, at $8 per ton, to $320,000. In Par. 20 the saving indicated in generation, $39,760, would be only 12.4 per cent of the fuel cost. The distribution saving would be 3.8 per cent, and the utilization saving 7.8 per cent of the fuel cost. The total saving would be 24 per cent of the fuel cost, in itself a notable achievement.

In finishing plants the power cost is usually about 25 per cent of the total operating expense, so that in this particular case the total expense would be approximately $1,250,000 per year. The various savings based on this total, then, would be: For generation, 3 per cent; for distribution, 0.95 per cent; for utilization, 1.95 per cent. The total gross saving, then, is 6 per cent of the total operating cost. These savings could not be accomplished without some expenditure, and it is unfortunate that the author has failed to include figures, both initial and continuing, of this expenditure. Assuming an initial expense of $55,000, the net saving the first year would be $21,000, or 6.7 per cent of the fuel cost, or 1.7 per cent of the total operating cost. There is no intention to belittle the work of the author, but it would seem advisable to consider all the relations in a paper of this character.

The writer believes that the summary of reports of the character outlined by the author should be comparative. The operations of the current month should be compared with the operations of the same month of the preceding year. Temperature conditions would be the same, and these have a marked effect on steam consumption, as we are usually unable to separate the steam for heating from that used for process. The comparison of similar months in succeeding years eliminates this cause of variation. Such a comparison should also show the change between this year and last, and should show the percentage of such change. It should also be cumulative and sum up the months of the fiscal year to date. One advantage of this is that the twelfth monthly report will be the yearly total, and it will be possible to ascertain whether or not the fuel paid for checks with that reported by the power plant as having been used.

Daniel M. Bates. In the finishing industry it is more important to cut down costs than to raise prices, and the writer knows of no way of doing this that is comparable with the opportunities in the power plant. At the Lewiston Bleachery and Dye

Works there were a number of boilers that were nearly worn out, and that were driven very hard. The firm of Day and Zimmerman, called in as consulting engineers on the question of power, discovered peak loads thrown on the boilers twice each day, due to the kiers being put on the line one after the other. A charting of the load showed that it was possible to so plan the work of the kiers that the load due to them was distributed over the twenty-four hours, with a resulting smoothing out of the boiler load curve, which enabled the works to keep the boilers in service for six or eight years longer.

In Par. 23 the author refers to the saving effected by the simple shutting off of pipe lines over week ends. The writer has found it to be of the greatest possible importance to get out into the plant when it is not running in order to discover wastes of steam. In this way he has found valves leaking steam into the cans when they were shut down over night, so that they were half full of condensate in the morning.

In Par. 26 the author mentions the use of low-pressure steam to bring water to the boiling point. At Lewiston, great economies were effected by using exhaust steam to heat a big feedwater tank for use in kier boiling. The steam would otherwise have been wasted, and there was always an ample supply of hot water for the kiers when it was needed. There is always an enormous waste of heat in the waste liquor from the kiers. A number of attempts were made to recover this heat but the expedient of pumping the liquor from one kier to the next one was finally adopted, adding sufficient caustic or other chemical to bring the liquor up to strength. The liquor was taken from the top of the kiers, so that the sludge would remain at the bottom. All the heat that was in the liquor was thus saved, representing a coal saving of two or three tons per day.

Geoffrey C. Brown. The author ably sets forth the economies effected in a *large* power plant by better management. The writer recently has had a practical demonstration of the fact that proportional economies may be effected along similar lines in a *small* power plant.

About two years ago the Jacques Kahn Mirror Company undertook a complete reorganization of its New York factory, including the small power plant. The latter included two 200-hp. hand-fired boilers supplying steam to two turbo-generators, in addition to providing live steam for power-plant auxiliaries and use in manufacturing processes. The turbine exhaust was used in winter for heating the factory and in summer was wasted.

While the power-plant equipment was fairly adequate to produce its quota of steam and power, the necessity for managing it, in the modern significance of the term, had never been realized. Records which might have indicated operating efficiency were conspicuously absent. A calculation showed an approximate

production of six pounds of steam per pound of coal burned — an evaporative efficiency of only 49 per cent. A crude weekly log report showed the daily output of electric power from the generators without any record of steam production. The unit costs of electric power per kilowatt-hour or of steam per 1000 lb. were not known, nor had the amounts of live steam or electric energy consumed by individual factory departments ever been either metered or estimated. Power-plant supplies were purchased on a verbal request from the power-plant engineer and were used by him without record. No inventory of fuel or supplies had ever been maintained, and power-plant costs were roughly visualized once a month by the coal transactions appearing on the company's general books. The plant obviously provided a promising field for investigation. The conditions described are reasonably typical of those existing in the majority of small factory power plants.

As the result of an investigation, the following changes were made:

a Republic steam flowmeters recording from each of the two boilers were installed in the boiler room. These enable the fireman to see at a glance whether his ratios of draft and steam are correct, and to regulate accordingly either the amount of coal fired or the volume of draft maintained.

b Watt-hour meters and St. John steam flowmeters were installed to measure the actual amounts of electric power and process steam used by each factory department.

c The power plant was brought within the scope of a newly organized factory planning department to which the power-plant engineer was required to send a weekly report abstracted from his daily power-plant log. This showed the amounts of fuel and power-plant supplies used, and the daily details of plant performance. With the power plant under the jurisdiction of the factory planning office the fundamentals of storekeeping and cost finding, originally lacking, were extended to it.

The economies resulting from these changes are now being realized. The installation of the flowmeters in the boiler room increased evaporative efficiency at least 30 per cent. The aggregate saving resulting from the entire power-plant investigation was about $9000, or 35 per cent of the total power cost during the previous year. And of greatest value has been the important general result of placing the power plant under centralized control as definite as that exercised in planning production in the factory.

Prior to the investigation the directors shared the popular delusion that because a plant is small the expense of installing better methods would more than counterbalance any advantages derived therefrom. After a year of more economical operation, however, they are thoroughly satisfied with the new methods.

Reynolds Longfield.[1] The writer had a similar experience to that of the author, in the power plant of the Celluloid Company at Newark, N. J., at the time that Mr. Polokov was engaged there in an advisory capacity.

When the work was started the equipment was in bad condition. It was put in first-class shape and the employees taught the proper methods to use for best results. A quality task was then set and a bonus paid to employees for its proper performance. This task, with a bonus for its accomplishment, not only maintained efficient operation of the plant, but at the same time performed another and even greater service. It insured maintenance of the equipment. The men themselves were vitally interested in the upkeep. The management, which made it a point to investigate a lost bonus, that represented a loss to the company also, due to the falling down in plant efficiency, saw the necessity for immediate repairs.

The writer believes that Task and Bonus is as essential in the manufacture of power as it is in the manufacture of any commercial article.

Horace G. Killam.[2] Two obstacles always confront a power-plant engineer in a finishing plant. First, the production of power is of minor importance as compared with the production of cloth. Generally, investment in finishing machinery can be made to show a much greater return than the same investment in power-plant equipment.

In the second place, the power engineer cannot be expected to know the details of each finishing process. He is handicapped in attempts to effect steam economies by the fear of department heads that the quality of the product will suffer. The author has shown the possibilities of appreciable savings in that department which too often is considered merely a painful necessity.

The author, however, does not make clear the relative unit cost of live and exhaust steam. The writer's company is attempting to solve this question, possibly on a rather theoretical basis. They use, in certain processes, live steam at 100 lb. pressure and wherever possible exhaust steam at 15 lb. pressure, but charge for both at the same rate per 1000 lb. Since the B.t.u. content of 15-lb. steam is only two or three per cent less than that of 100-lb. steam, should not the cost per 1000 lb. show the same ratio?

It is realized that the apportionment of boiler-room cost between live steam, exhaust steam, and power offers an unlimited field for debate, and in the end the decision must rest on somewhat theoretical grounds. But after all, in the distribution and utilization of steam, we are dealing with energy. In the case of steam this is measured in B.t.u., and the writer believes that the relative

[1] U. S. Finishing Co., Providence, R. I.
[2] Mechanical Engineer, U. S. Finishing Co., Providence, R. I.

charge for live steam and exhaust steam should bear a close relation to the relative B.t.u. content.

Wallace Clark. The usual manufacturer considers the power problem solved when he has reduced the cost of generating power through the introduction of scientific methods. The author, however, demonstrates that this is only the beginning of economies. This new field demands knowledge of at least three phases of management, i.e., the processes in which steam is used, shop accounting to allocate accurate costs to the various grades and styles of products in order to secure business at a profit, and planning the utilization of steam. A point that the author did not mention is that the more closely plans for the use of steam are synchronized with plans for routing work through the plant, the greater will be the saving. The author has opened up what is apparently a fertile field in the reduction of waste, not only in the textile industry but in the manufacture of paper, rubber, steel, and other products.

The Author. In reply to Mr. McGregor, the author would state that the cost of the meters and the professional services required to check up the records made by the engineering department was about $15,000. The cost of the piping cannot be given definitely, as the piping changes occurred in a reconstruction period and a great deal of the piping cost would have been incurred whether the system outlined in the paper had been developed or not. While we have no definite figures on the piping, we felt that an expense of even 20 per cent of the total cost of total steam generated would not be excessive.

The author agrees with Mr. McGregor that fuel costs should be adjusted on the basis of monthly cumulative costs instead of on the basis of the fiscal year with an extra week left out, allowing for the neglect of the accountant to obtain the actual weight of fuel used. There are many chances for error in the usual system of accounting, and it is only by eternal vigilance that the accounting department is enabled to give us true costs.

Mr. Bates has stated truly that it is necessary to go into the plant when it is shut down, in order to detect wastes of steam. A case in point is a leakage of a 2-in. valve in one of our dye houses which was permitted to run for a period of over an hour one night. This leakage was picked up by the electrically operated instrument, three-eighths of a mile distant, unknown to the department foreman. In regard to waste liquor, we have attempted to build an economizer to recover the heat, but we have found up to the present time that the chemical end of the problem in preventing the acid, etc., in the liquors from destroying the economizer material, is quite difficult of solution. Waste keir liquors have been recovered by filtering and then pumping back into the keirs. The saving due to the recovery of waste liquor probably amounts to

about $100 a day. A similar study is now being carried on in regard to the recovery and re-utilization of such chemicals as caustics.

In regard to the cost of live and exhaust steam as stated by Mr. Killam, we arrived at the relative costs through data collected by test meters. These data covered the amount of steam going through the engine, and the loss in radiation and in the condenser. We feel that our results are much more accurate than if we charge exhaust and live steam at the same price. One reason for a different charge for the two kinds of steam is that it furnishes an incentive for the department head to accomplish the work of his department at a lower cost and thereby increase the bonus for production. The use of exhaust steam will naturally lower the costs, and it is well to furnish an incentive for its use rather than to put it through the exhaust head.

No. 1906

A GRAPHICAL STUDY OF JOURNAL LUBRICATION

By H. A. S. Howarth,[1] Philadelphia, Pa.

Member of the Society

This paper visualizes the characteristics of the oil film and the pressures within it for a journal completely surrounded by its bearing. The influences of clearance and viscosity upon the journal friction are quantitatively shown by means of a chart that can readily be used for designing bearings. Several interesting examples are solved by means of this chart.

A few of the characteristics of the journal partially surrounded by the bearing are also shown by curves. The study of partial bearings is not yet completed.

THE hydrodynamic theory of lubrication, first developed by Osborne Reynolds and subsequently simplified by A. Sommerfeld and W. J. Harrison, forms the basis of this exposition. An attempt is made to express the theory in a way that will be easily applicable to the solution of practical problems. Much of the early mathematical treatment of the subject looks formidable, and therefore does not encourage general study. Fortunately, W. J. Harrison, Fellow of Clare College, Cambridge, England, has presented the subject in simple mathematical form in his paper The Hydrodynamical Theory of Lubrication . . ., which appeared in the Transactions[2] of the Philosophical Society, Cambridge, in 1913. His formulas are given below and are used for plotting the curves that follow.

2 The case of a journal completely surrounded by its bearing will be considered first. Fig. 1, which conforms with Harrison's diagrams and symbols, shows the journal supporting the bearing. It is assumed that the journal bearing is so long that the effect of end leakage of oil is negligible. The journal revolves in the direction of the arrow with a surface velocity U. The point of nearest approach will be on a diameter perpendicular to the line of the load R, provided the oil film is continuous and completely fills the space between the journal and bearing. This has been demonstrated by both Sommerfeld and Harrison. a = radius of the journal whose center is at O. $a + \eta$ = radius of the bearing

[1] Chief Engineer, Kingsbury Machine Works.

[2] Vol. xxii, no. iii, pp. 39–54.

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whose center is at O'. h = thickness of the oil film at angle θ from diameter through OO' and the point of nearest approach. The eccentricity OO' is expressed in terms of the radial clearance by using a factor c; thus $c\eta = OO'$. This factor c will be zero when O coincides with O'. It will be unity when the eccentricity equals η and the journal touches the inside of the bearing.

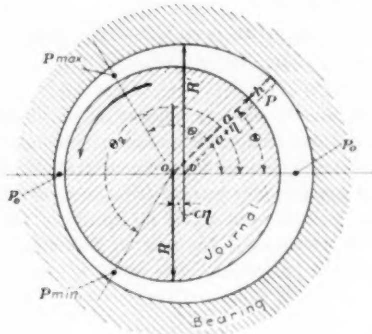


FIG. 1 JOURNAL SUPPORTING THE BEARING. THE JOURNAL PRESSES UPWARD AND THE REACTION R OF THE OIL-FILM PRESSURE IS THEREFORE DOWNWARD

3 The film thickness h can be expressed quite simply by employing an approximation that is justified by the customary ratios of η/a used in practice. It is then evident from Fig. 1 that

$$h = \eta(1 + c \cos \theta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

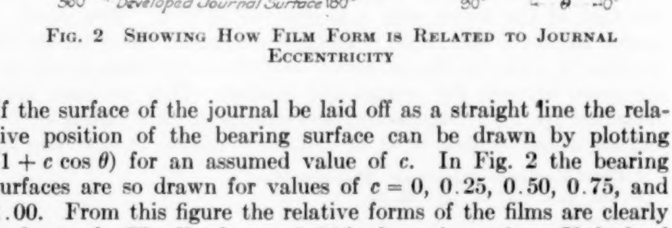


FIG. 2 SHOWING HOW FILM FORM IS RELATED TO JOURNAL ECCENTRICITY

If the surface of the journal be laid off as a straight line the relative position of the bearing surface can be drawn by plotting $(1 + c \cos \theta)$ for an assumed value of c. In Fig. 2 the bearing surfaces are so drawn for values of $c = 0$, 0.25, 0.50, 0.75, and 1.00. From this figure the relative forms of the films are clearly understood. The film for $c = 0.50$ is shown in section. If the load R is zero, c will be zero, and the film will be uniform in thickness

because the centers of the journal and bearing coincide. As the load increases the eccentricity also increases, until with infinite load the surfaces touch and $c = 1$.

4 Referring again to Fig. 1, the pressure that is developed within the film varies from P_0 when $\theta = 0$, through P_{max} at θ_1, P_0 again when $\theta = 180$ deg., P_{min} at θ_2, and finally to P_0 when $\theta = 360$ deg. The angles of maximum and minimum pressure (θ_1 and θ_2) are found from the formula

$$\cos \theta_1 = \cos \theta_2 = -\frac{3c}{2 + c^2} \quad \text{. . . . . . .} \quad [2]$$

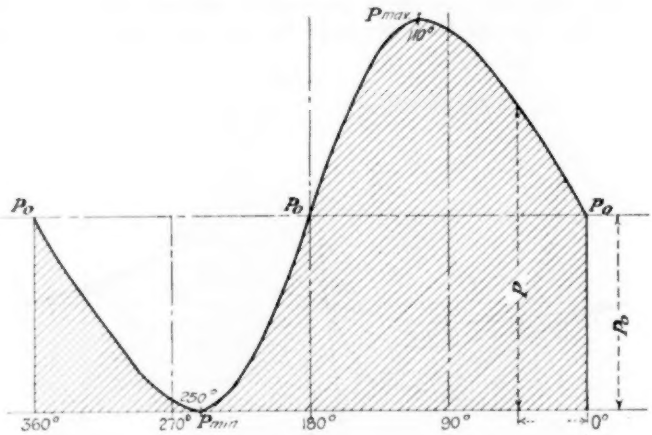


Fig. 3 Pressure Diagram for $c = 0.2$

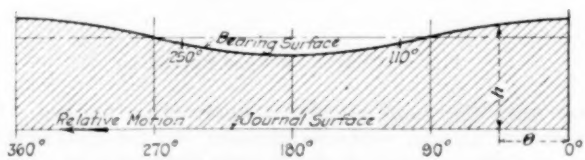


Fig. 4 Film-Thickness Diagram for $c = 0.2$

Hence the positions of the points of maximum and minimum pressure are symmetrical with respect to diameter OO'; or $180° - \theta_1 = \theta_2 - 180°$.

5 The film pressure P for any value of θ is found from the formula

$$P = P_0 + \frac{6\mu U a}{\eta^2} \left\{ \frac{c \sin \theta \ (2 + c \cos \theta)}{(2 + c^2)(1 + c \cos \theta)^2} \right\} \quad . \ . \ . \quad [3]$$

The only new symbol in this formula is μ, the absolute viscosity of the lubricant. By making suitable assumptions for μ, U, a, η,

and c, the pressures relative to P_0 can be found for all values of θ and then plotted on a polar or a rectilinear diagram.

6 Fig. 3 shows a rectilinear pressure diagram for $c = 0.2$. In order to establish an axis of ordinates $P_{\min}$ is assumed to be zero.

7 Directly below Fig. 3 is Fig. 4 with the same scale of abscissas. It is similar to the curves of Fig. 2 but is plotted for $c = 0.2$. It shows the relative film thickness corresponding with each value of θ. Figs. 3 and 4 together show how the pressure varies with relation to the film thickness. Beginning at 0 deg., where the film is thickest, the pressure increases from P_0 while the film gets thinner, until the maximum pressure is reached when θ is 110 deg. The pressure then decreases but the film continues to get

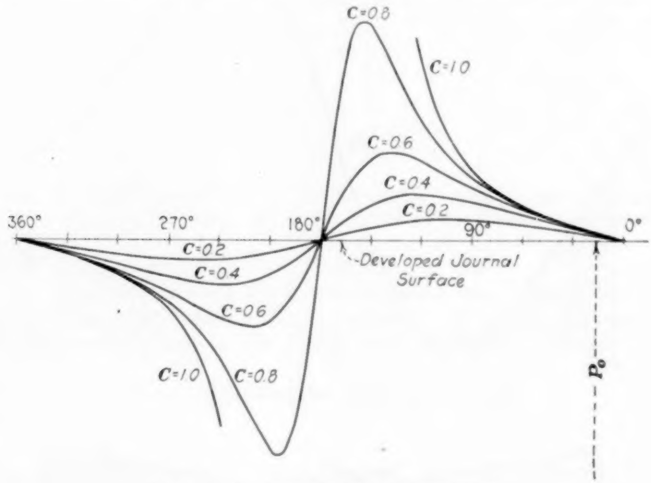


Fig. 5 Curves of Film Pressures

thinner until the point of nearest approach is reached. From this point on the film thickness increases, but the minimum pressure is not reached until θ is 250 deg. From this point on the film thickness and the pressure increase together until the circuit is completed.

8 The angle of maximum pressure θ_1, and also the maximum pressure itself, increases with c. In Fig. 5 the pressure curves relative to P_0 are drawn for $c = 0.20$, 0.40, 0.60, and 0.80. The scale of ordinates used is different from that employed in Fig. 3, and the zero pressure line was not drawn because a different one would be required for each value of c if assumed to coincide with $P_{\min}$.

9 The pressures in the film when integrated parallel to the applied load may be represented by R'. This integration may be performed graphically as indicated in Fig. 6. For this purpose

the ordinates of Fig. 5 were laid off vertically, above and below the P_0-axis of Fig. 6, the spacing being obtained by projecting the angular positions. The area enclosed by each curve is proportional to the load carried by the journal for the assumed value of c. These curves are drawn for $c = 0.2$, 0.4, 0.6, and 0.8.

10 It was stated in Par. 2 that the point of nearest approach was on a diameter perpendicular to the load. When the pres-

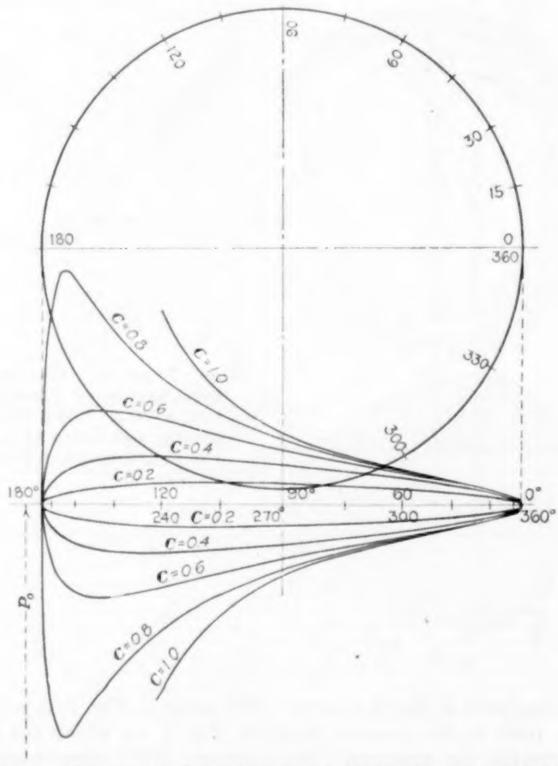
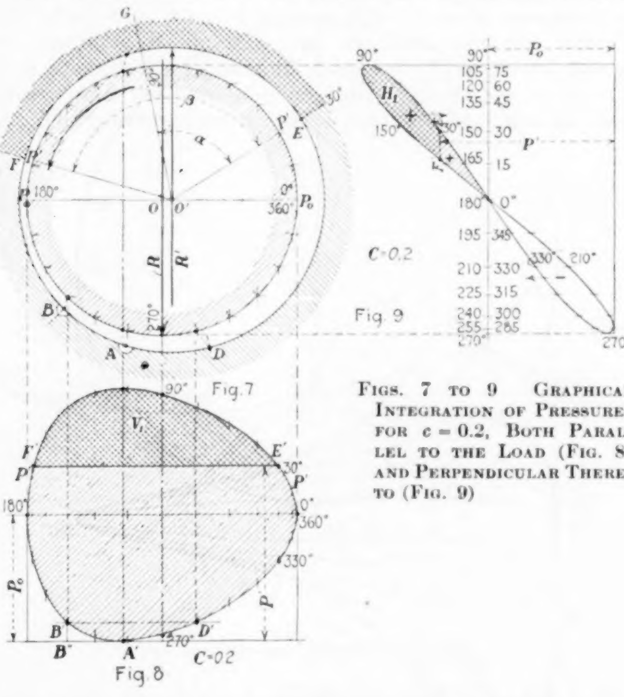


Fig. 6 Graphical Integration of Pressures in the Film
(The enclosed areas are proportional to the carrying power of the film.)

sures are integrated in a direction perpendicular to this diameter, i.e., parallel to the load, they have the maximum value for any given value of c. An integration at right angles to the load equals zero. For this position there is therefore equilibrium. The effect of the friction forces is so very minute compared with that of the pressures that it is negligible so far as this equilibrium is concerned. In Figs. 7, 8, and 9 the graphical integration of pressures is shown for $c = 0.2$ both parallel to the load (Fig. 8) and perpendicular thereto (Fig. 9).

11 Referring to the pressure formula, Equation [3], it is evident that P_0 can have any value that will satisfy practical considerations. If an oil inlet be provided in the bearing of Fig. 7 at the point of minimum pressure A, oil may be introduced there at atmospheric pressure, or at some other pressure determined by the oiling device. The carrying capacity of the oil film is independent of this inlet pressure so long as the film remains complete.



FIGS. 7 TO 9 GRAPHICAL INTEGRATION OF PRESSURES FOR $c = 0.2$, BOTH PARALLEL TO THE LOAD (FIG. 8) AND PERPENDICULAR THERETO (FIG. 9)

If the oil groove is placed at some other point B, Fig. 7, it is necessary to refer to the pressure diagram, Fig. 8, on which the point B' represents the pressure. The ordinate $B'B''$ when measured by the pressure scale must be equal to or greater than the supply pressure plus absolute atmospheric pressure, because the minimum pressure at A' cannot be less than absolute zero pressure.

12 Additional formulas given by Harrison for the journal completely surrounded by its bearing are given below. The resultant R of the integration of the film pressures on the journal and in the direction of the load, is given by Equation [4] for one unit of length of the bearing.

$$R = \frac{12\pi\mu U a^2}{\eta^2}\left\{\frac{c}{(2+c^2)(1-c^2)^{\frac{1}{2}}}\right\} \quad . \; . \; . \; . \quad [4]$$

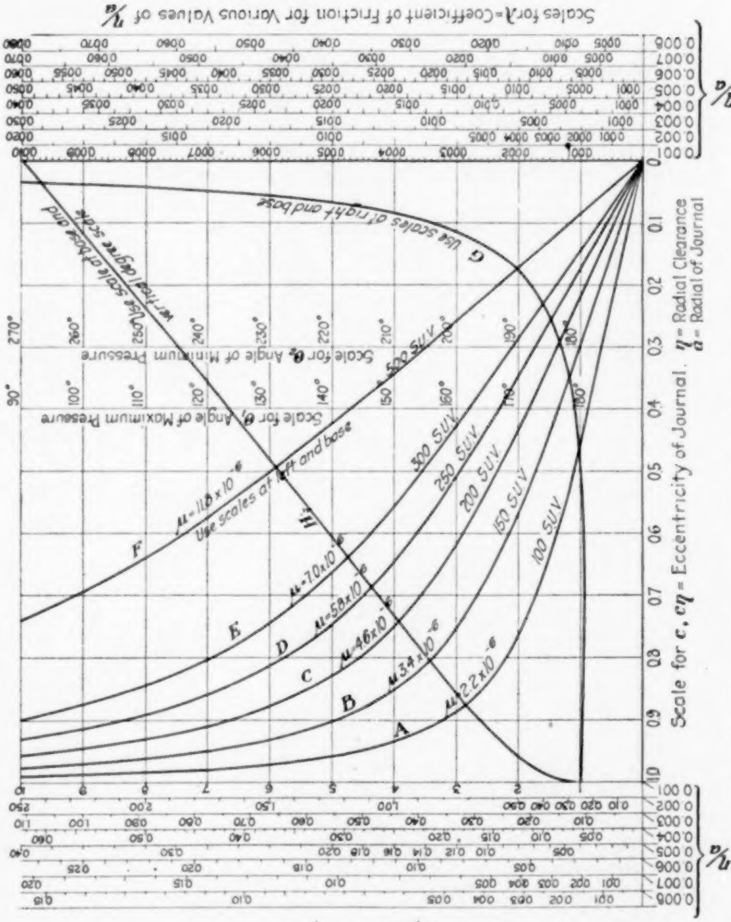


Fig. 10 Characteristics of Journal Surrounded by Bearing

($c\eta$ = eccentricity of journal; η = radial clearance; a = radius of journal. For interpretation of symbols, see Fig. 1.)

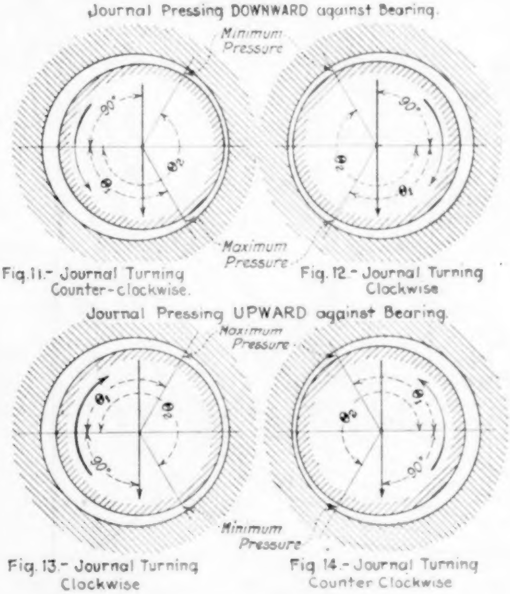
13 The moment M of the friction force *on the journal* is given by Equation [5] for each unit of bearing length.

$$M = \frac{4\pi\mu Ua^2}{\eta}\left\{\frac{1+2c^2}{(2+c^2)(1-c^2)^{\frac{1}{2}}}\right\} \quad . \quad . \quad . \quad . \quad . \quad [5]$$

14 The friction coefficient λ *for the journal* is given by Equation [6].

$$\lambda = \frac{M}{Ra} = \frac{\eta(1 + 2c^2)}{3ac} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

15 In order that Equations [3], [4], and [5] may be readily applicable to the solution of bearing problems it is necessary to rewrite them, converting U into r.p.m. $U = 2\pi aN/60$. It is also advisable to keep the ratios (a/η), or (η/a), together. Remembering that $R = 2aw$, w being the mean pressure per square



FIGS. 11 TO 14 SHOWING HOW ANGLES θ_1 AND θ_2 SHOULD BE MEASURED WITH RELATION TO DIRECTION OF ROTATION AND DIRECTION OF PRESSURE OF JOURNAL UPON BEARING

inch of projected area, as regularly used in bearing design, [4] can be rewritten as follows:

$$\frac{w}{N} = \frac{\pi^2}{5}\,\mu\left(\frac{a}{\eta}\right)^2 \left\{\frac{c}{(2 + c^2)(1 - c^2)^{\frac{1}{2}}}\right\} \quad . \quad . \quad . \quad . \quad [7]$$

Equation [6] for the *journal friction coefficient* may be rewritten as shown in [8], keeping the ratio (η/a) apart from the term containing c.

$$\lambda = \left(\frac{\eta}{a}\right) \left\{\frac{1 + 2c^2}{3c}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

16 Equations [2], [7], and [8] appear to be the most useful for designing full bearings of this kind. A study of them is ex-

pressed graphically in the chart Fig. 10, from which the journal characteristics can be very easily determined. Curve *G* was plotted from Equation [**8**], assuming $(\eta/a) = 0.001$. Suitable scales for other values of (η/a) were then added. Curves *A*, *B*, *C*, *D*, *E*, and *F* were plotted from Equation [**7**], assuming $(\eta/a) = 0.001$, each curve having been drawn for the oil viscosity μ marked on it. The scales for other values of (η/a) were then added. The absolute viscosity is marked on each curve in English units — inches, pounds, seconds. The corresponding commercial viscosity as determined by a Saybolt universal viscosimeter (S.U.V.) is marked on each curve. The conversions were made from P. C. McIlheney's table as published in the *Journal of Industrial and Engineering Chemistry*, vol. 8 (1916), p. 434. A density of approximately 0.84 was assumed and the viscosities in centipoises were converted to English units by dividing by 6,900,000. For the range of viscosities illustrated in Fig. 10 the conversions from S.U.V. to absolute English units can be made with fair accuracy by means of the following formula:

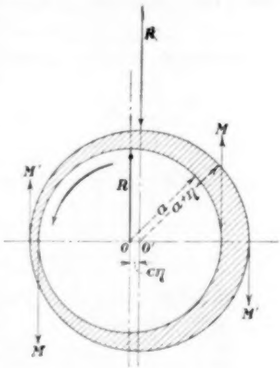


Fig. 15 Film Equilibrium Diagram

$$\mu = 0.0286 \times 10^{-6} \times \text{density} \times (\text{S.U.V.} - 8) \quad . \quad . \quad [\mathbf{9}]$$

17 Data on the location of points of maximum and minimum pressure can be also obtained from the chart Fig. 10. Curve *H* was plotted from Equation [**2**], using the *c*-scale at the base and the two angle scales for θ_1 and θ_2 laid off vertically near the middle of the figure.

18 Much interesting practical information on bearing problems can now be obtained from Fig. 10. The following examples show how it may be used. In order to avoid confusion when interpreting the readings from Fig. 10, reference may be made to Figs. 11, 12, 13, and 14, which show how the angles θ_1 and θ_2 should be measured with relation to the direction of rotation and the direction of pressure of the journal upon the bearing.

Example 1. Find the journal friction coefficient for a full bearing whose diametral running clearance is 0.001 in. per inch of diameter, if the speed is 400 r.p.m., the mean pressure 100 lb. per sq. in., and the oil viscosity is 100 sec. S.U.V.

First calculate $w/N = 100/400 = 0.25$. Then enter Fig. 10 at the left-hand vertical scale marked $(\eta/a) = 0.001$ and locate the point whose value is 0.25. From this point read horizontally to curve *A*, then down to the *c*-scale where we find $c = 0.12$, then up to curve *G* and horizontally to the λ scale corresponding with $(\eta/a) = 0.001$. A value of $\lambda = 0.0029$ is found.

In order to determine the positions of maximum and minimum film pressure, refer to Fig. 10 again. Enter its base with $c = 0.12$ and read up to curve H and across to the scales for θ_1 and θ_2. The angles are found to be as follows: $\theta_1 = 101$ deg., and $\theta_2 = 259$ deg.

EXAMPLE 2. What clearance will give the minimum friction for a journal revolving at 1200 r.p.m. if the unit pressure is 120 lb. per sq. in. and the oil viscosity is 150 S.U.V.?

This must be solved by trial, but takes very little time. $w/N = 120/1200 = 0.1$. Assuming $(\eta/a) = 0.001$, $\lambda = 0.010$; $(\eta/a) = 0.002$, $\lambda = 0.0057$;

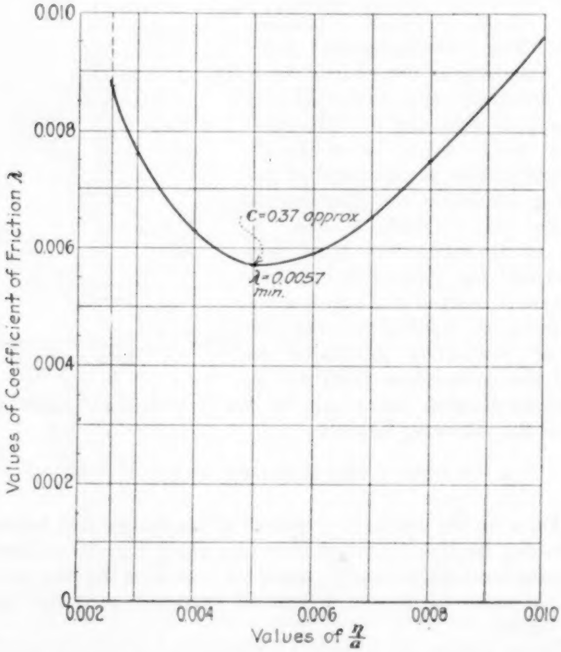


FIG. 16 VARIATION OF λ WITH (η/a) FOR A SPECIFIC CASE

[Example data: $N = 1200$; $w = 60$; $\mu = 3.4 \times 10^{-6}$ (S.U.V. = 150.)]

$(\eta/a) = 0.003$, $\lambda = 0.0043$; $(\eta/a) = 0.004$, $\lambda = 0.0041$; $(\eta/a) = 0.005$, $\lambda = 0.0048$; etc. Apparently the proper clearance is about $(\eta/a) = 0.004$. A similar problem in which $w = 60$ is plotted in Fig. 16, showing that λ min $= 0.0057$. Decreasing the unit pressure w is thus found to increase the friction coefficient.

19 Before leaving the discussion of the journal completely surrounded by its bearing, attention should be given to an interesting statement made by W. J. Harrison in his paper above referred to. Fig. 15 shows the forces that act on the oil film in the bearing and hold it in equilibrium as the journal rotates. Considered statically,

$R = R'$. The moments must balance also. Hence we have the relation in the formula

$$M = M' + Rc\eta \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad [10]$$

In this case M' is the moment of the friction between the film and the surrounding *bearing*, whereas M is the corresponding but greater friction moment between the film and the *journal*. The difference between these is obviously the couple RR' whose moment is $Rc\eta$. Mr. Harrison says that this difference would help explain the inconsistency between results of tests of bearings. Some experimenters

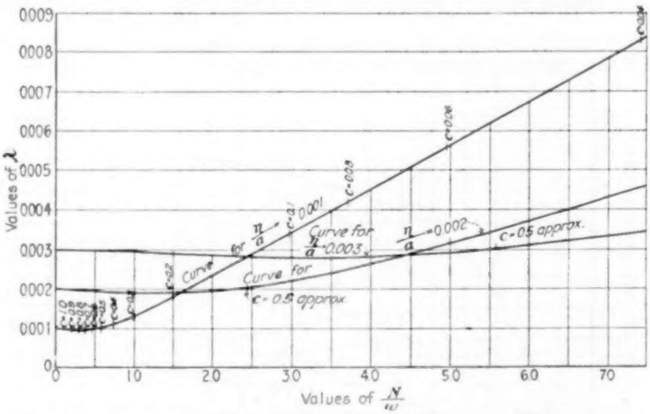


FIG. 17 VARIATION OF λ WITH (N/w), μ BEING TAKEN EQUAL TO 3.4×10^{-6}

measure the journal friction, but most of them measure the bearing friction,

$$M' = \frac{4\pi\mu U a^2}{\eta}\left\{\frac{(1-c^2)^{\frac{1}{2}}}{2+c^2}\right\} \quad . \ . \ . \ . \ . \ . \ . \quad [11]$$

The coefficient of friction λ' between the film and the bearing is given in Equation [**12**].

$$\lambda' = \left(\frac{\eta}{a}\right)\frac{1-c^2}{3c} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad [12]$$

20 W. J. Harrison examined the relation between M and M' and found as follows:

$c = 0$	0.1	0.4	0.6	0.9	1.0
$M/M' = 1$	1.03	1.51	2.69	13.8	∞

Evidently the ratio varies from 1 to ∞. For example, when tests are made in which the bearing surfaces are very close together at one side compared with the clearance, the friction moment on the journal is vastly greater than the friction moment on the bearing.

Journals Partially Surrounded by the Bearing

21 Tower ran his experiments on a bearing partially surrounding its journal. Hence when Professor Reynolds made his analysis of these experiments he investigated the case of a journal partially surrounded by its bearing. His work, as previously stated, is not simple enough to be easily used by others. Sommerfeld later made an analysis of a journal completely surrounded by its bearing and also of one in which the bearing extended exactly half-way around. W. J. Harrison later solved the problem of the complete bearing in a simple manner which can be readily used. Harrison's work

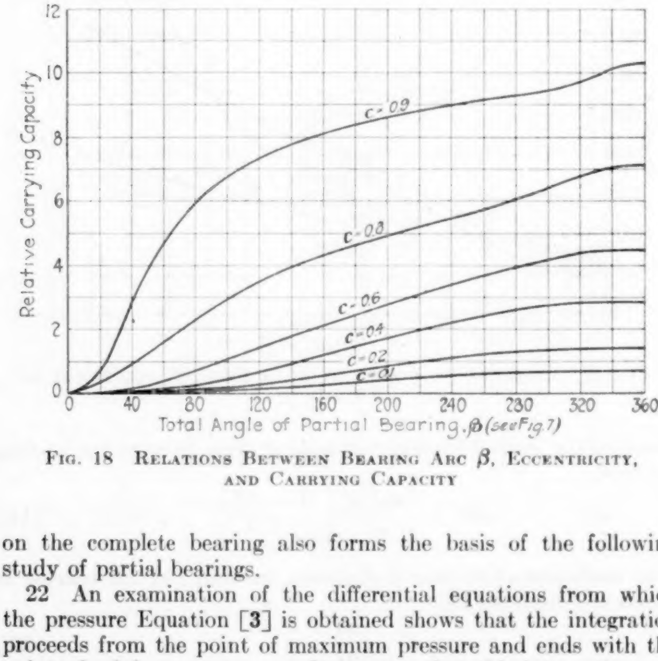


Fig. 18 Relations Between Bearing Arc β, Eccentricity, and Carrying Capacity

on the complete bearing also forms the basis of the following study of partial bearings.

22 An examination of the differential equations from which the pressure Equation [**3**] is obtained shows that the integration proceeds from the point of maximum pressure and ends with the point of minimum pressure. It appears that this integration can be stopped off anywhere around the film provided only that the pressures at the two ends of the film are such as can readily exist in a bearing. The simplest case is where the pressures are the same at both ends of the film and are atmospheric.

23 When dealing with partial-bearing films the conditions of equilibrium must be carefully examined to determine the direction of the resultant pressure upon the bearing. The process by which partial bearings can be studied will be evident from the following comments upon Figs. 7, 8, and 9. It may be assumed that the

bearing begins where $\theta = 30$ deg., i.e., at point E in Fig. 7, and that the pressure at that point is P'. We must first determine the next point in the direction of rotation of the journal in which the pressure is the same as at 30 deg. Referring, therefore, to the points E' in Figs. 8 and 9, corresponding with point E in Fig. 7, the lines $E'F'$ may be drawn parallel with the zero-pressure axis and cutting the closed pressure loop at F'. The pressure at both E' and F' will be P'. The point F' in Fig. 8 or 9 may be transferred so as to locate point F in Fig. 7. This may therefore represent the end of the partial bearing that begins at E which is 30 deg. from the diameter that is perpendicular to the load. The angular length of the partial bearing is β.

24 If we shade again the portion of the pressure loop lying above the line $F'E'$ in Fig. 8, this double-cross-hatched section

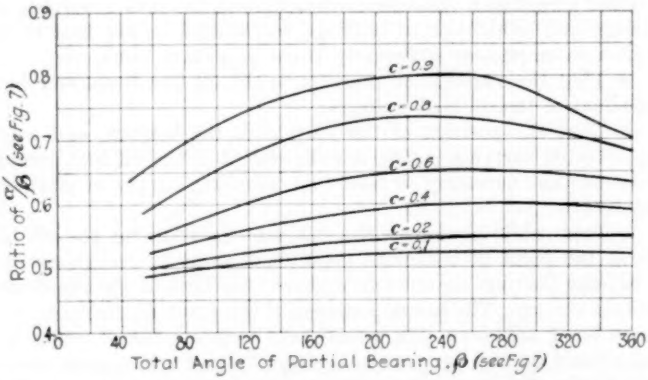


Fig. 19 Location of Applied Load with Relation to Bearing Arc β

will represent the vertical component V_1 of the film pressure between the partial bearing EF and the journal.

25 Referring now to Fig. 9, if we again cross-hatch the portion of the loop at the left of $E'F'$, this double-cross-hatched section represents the horizontal component H_1 of the film pressures between the partial bearing EF and the journal. This acts to the right on the journal and to the left on the bearing.

26 The relative magnitudes of V_1 and H_1 can be determined by measuring the double-cross-hatched areas with a planimeter. The resultant of these two will act along the line $O'G$ in Fig. 7 at angle α from $O'E$. This line of resultant pressure will divide the partial bearing β into two portions, α and $\beta - \alpha$, the portion α representing the leading portion and the remainder the trailing portion when the bearing is considered with reference to the direction of rotation of the opposing journal surface.

27 The method outlined above has been applied to the exami-

nation of partial journals varying in arc from 0 deg. to 360 deg. for various values of the eccentricity c. The results thus far obtained are indicated by Figs. 18 and 19; the first showing the relation between bearing angle, eccentricity, and carrying capacity; the second the location of the applied load with relation to the bearing arc β.

28 The investigation of partial bearings is being continued and it is expected that the results can be charted in as useful a manner as those in Fig. 10. This work will probably be completed before the coming spring.

DISCUSSION

Daniel P. Barnard, 4th,[1] and E. V. Murphree.[2] While the author furnishes a valuable contribution to the laws governing the design and lubrication of bearings, the writers believe that he has failed to emphasize sufficiently three important facts, viz:

a That the variable oil pressure in the oil film is entirely independent of the oil feed pressure;

b That the position of the journal in the bearing, and consequently its carrying power, is independent of the oil feed pressure beyond that necessary to insure an unbroken supply of lubricant; and

c That while obviously the optimum position for the oil hole is at the point of minimum pressure, it must be recognized that ordinary bearings do not follow closely the laws of the two-dimensional theory. The actual location of the point of minimum pressure being unknown, it is usually best to locate the oil supply hole at a point on the low-pressure side in line with the application of the load. This position usually deviates but little from that of the actual point of minimum pressure.

While attempts are often made to increase the carrying capacity of a bearing by increasing the oil supply pressure, the former is determined entirely by the internal operating conditions of the bearing, i.e., $(\mu U/R)$, (η/a), in the author's notation.

A two-dimensional consideration of the problem is of value to an understanding of the mechanism by which fluid-film lubrication occurs, but it leads to some erroneous conclusions of a serious nature. In the two-dimensional theory the distribution curve for the variable oil-film pressure must be symmetrical to the displacement axis, and therefore the direction of displacement of the journal must be at right angles to the applied load. This leads to the conclusion that if the actual minimum pressure (P_0-P) should tend to drop below a value of zero absolute, a rupture of

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[2] Assistant, School of Chemical Engineering Practice, Massachusetts Institute of Technology, Cambridge, Mass.

the film must occur. Such a rupture may be understood as a transition point from fluid-film to absorbed lubrication. The writers have a considerable amount of experimental evidence where fluid-film lubrication was obtained under conditions which, according to the two-dimensional concept, would involve negative pressure. These are shown in Figs. 20 and 21, which compare the observed friction of the bearings with that calculated by the equations given by the author. In Fig. 20 an apparent minimum pressure of -27 lb. per sq. in. exists just before the point of rupture is reached, while in Fig. 21 the minimum pressure is -154 lb. per

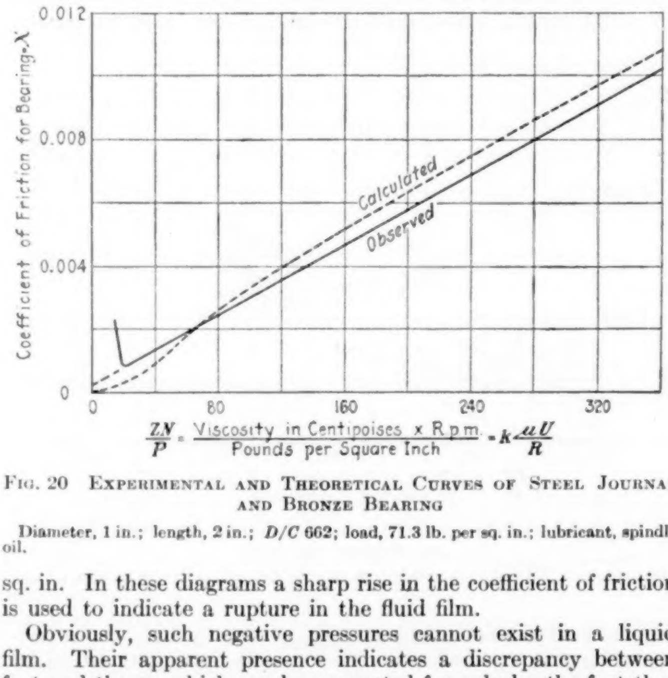

FIG. 20 EXPERIMENTAL AND THEORETICAL CURVES OF STEEL JOURNAL AND BRONZE BEARING

Diameter, 1 in.; length, 2 in.; *D/C* 662; load, 71.3 lb. per sq. in.; lubricant, spindle oil.

sq. in. In these diagrams a sharp rise in the coefficient of friction is used to indicate a rupture in the fluid film.

Obviously, such negative pressures cannot exist in a liquid film. Their apparent presence indicates a discrepancy between fact and theory which can be accounted for only by the fact that there actually exists a pressure variation along the length of the bearing. The difficulties attending three-dimensional treatment of a bearing are great. However, the writers have carried out the analysis to a point which indicates several important facts as follows:

a The distribution curve for the variable pressure is not symmetrical to the displacement axis

b For a short bearing (length not exceeding several diameters), the displacement of the journal is not at right angles to the load. This position, however, is approached at high values of $\frac{a^2}{\eta^2}\frac{\mu U}{R}$

c The maximum positive variable pressure in the oil film is ordinarily much greater than the corresponding negative pressure.

Conclusion *c* accounts for the possibility of existence of "negative" pressures in an ordinary bearing and at the same time precludes any chance of predicting by calculation the point of film rupture. The truth of this conclusion is obvious, since the ends of the conventional bearing are exposed to atmospheric pressure and the existence of a pressure below this figure will cause a flow inward toward the center of the bearing.

Guido H. Marx. The writer believes that the work of Sommerfeld and Harrison, upon which the author has based his paper,

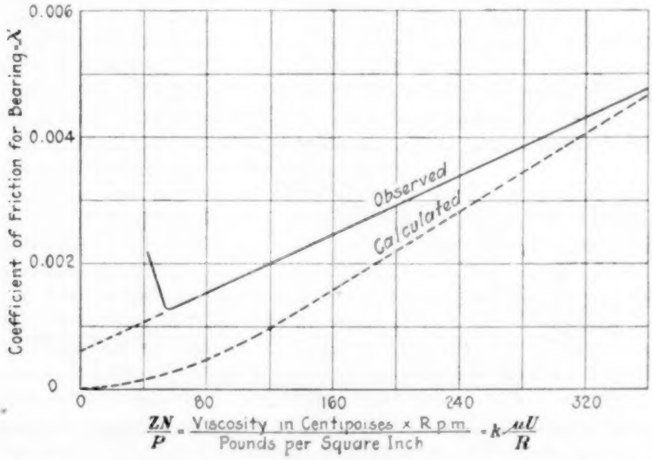

Fig. 21 Experimental and Theoretical Curves of Steel Journal and Bronze Bearing

Diameter, 1 in.; length, 2 in.; D/C 662; load, 71.3 lb. per sq. in.; lubricant, spindle oil.

forms a mathematical structure unsupported by sufficient experimental data as a foundation. Their results apparently are based on assumptions to some extent contrary to observed facts.

Three separate theses have been written under the writer's supervision by graduate students in mechanical engineering at Stanford University, covering investigations of the oil films of complete cylindrical bearings. That by B. M. Green was awarded the Society's student prize in 1916. He used ring oiling, and the point of nearest approach was in no case at right angles to the direction of pressure. In 1917–1919, Messrs. Bennett and Rifenberick continued and extended the investigation, using flooded lubrication with a head of oil above the journal. In no case did the point of nearest approach lie at right angles to the line of pressure, and in every case the center of the journal was lifted

above the center of the bearing, while the load was directed vertically downward. The same phenomenon was observed by Franklin and Hartmann in continuing the investigation last year. The writer does not attempt to explain it, but only reports it as an observed fact which seems to make untenable the fundamental assumptions of Harrison.

It should also be noted that a Russian, Petroff, developed the hydrodynamic theory of friction independently of and coincidently with, if not prior to, Osborne Reynolds. He should at least receive

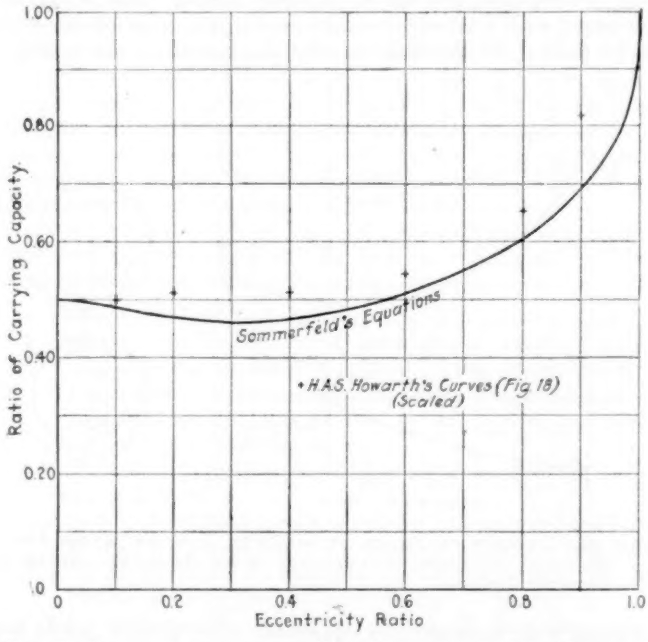

FIG. 22 CARRYING CAPACITY OF HALF BEARING AND FULL BEARING — IDEAL LUBRICATED JOURNAL

joint credit. The writer is not casting doubt upon the hydrodynamic theory, but is merely suggesting that certain physicists and mathematicians in applying it have overlooked some of the complexities in the phenomena and too greatly simplified their conclusions.

A. R. ROBERTS. While the bearing to which the author's curves apply is an ideal one, in that no account is taken of distortion due to load and the surfaces are assumed to be perfectly true and smooth, the assumptions made are not such as to modify to a great extent the conclusions which may be drawn as to the characteristics of real bearings. It is doubtful, however, whether in a

practical bearing, even with an ample supply of lubricant available, there exists the homogenous film assumed in the ideal case. The paper refers to the possibility of sub-atmospheric pressures existing in the film, and with such pressures air and other gases may separate from the liquid, and conceivably the film be ruptured. Such a condition might help to account for some of the difficulties of checking experimentally the results of mathematical analysis.

The case of partial bearings is perhaps of greater practical importance than that of full bearings. The particular case of a half bearing has been solved by Sommerfeld. His results are compared with those of the author in Fig. 22. The full-line curve is the ratio of the carrying capacity of a half and a full bearing as

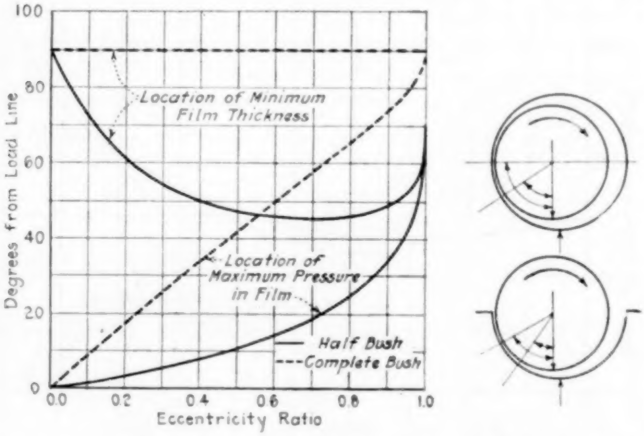

FIG. 23 LOCATION OF POINTS OF MINIMUM FILM THICKNESS AND OF MAXIMUM PRESSURE IN FULL AND HALF BEARINGS — IDEAL LUBRICATED JOURNAL

derived from Sommerfeld's equations. The plotted points were obtained by scaling the author's Fig. 18. The agreement is fair and while the possible errors in scaling are large, the discrepancies appear to be due to an assumption by the author.

The nature of this assumption may be realized by a study of Figs. 7 and 23. In a full bearing, as the load is applied the axis of the journal moves relatively to the axis of the bearing in a straight line along the diameter at right angles to the load line, and the point of minimum film thickness is always at the end of this diameter, i.e., at 180 deg. in Fig. 7. In the half bearing, according to Sommerfeld, the axis of the journal moves in a curved path so that the point of minimum film thickness lies between 135 and 180 deg., depending upon the eccentricity. The position of this point and of the point of maximum film pressure is not the same for the half bearing as it is for the full bearing as shown by Fig. 23.

Following the author's method, apparently the case of the half bearing can be worked out by cutting the full bearing in Fig. 7 along the horizontal diameter, removing the lower half, and leaving the pressure distribution and relative position of the axes unaltered. It would seem, therefore, that the author's values for carrying capacity should differ somewhat from Sommerfeld's. As the difference is not very serious, the value of the author's work on partial bearings is not appreciably diminished.

M. D. Hersey. The reduction of the hydrodynamic theory of lubrication to actual curves and every-day units permits its ready application to practical problems of design. A further result of the author's paper is the putting of the theory in a more usable form for research. For instance, by means of Fig. 5, we can now investigate the variation of viscosity in different parts of the film, assuming an experimental value for the increase of viscosity with pressure, and thus check the reasonableness of the assumption of uniform viscosity on which the theory is based.

Some of the particulars in which the classical theory probably differs from the true physical facts may be emphasized. These include the following basic assumptions:

a Uniform viscosity
b Freedom from leakage of oil at ends, and maintenance of film pressures shown by Fig. 5 clear up to the ends of the bearing, where it must then be assumed that these local pressures are abruptly reduced to the surrounding pressure P_0
c Geometrical perfection of bearing surfaces, freedom from distortion, etc.
d Absence of external couple about an axis perpendicular to that of the journal, such as might be caused by belt pull
e Rate of oil feed low enough to eliminate thrust due to change of momentum of the entering oil stream
f Film thick enough to possess the ordinary bulk properties so that no account need be taken of unknown "oiliness" factors.

While these, and other assumptions that may be involved, are doubtless understood by the author and research workers in general, they may be overlooked by certain classes of readers. Fortunately, these assumptions are substantially correct under ordinary conditions as proved by Harrison in the paper cited, where he refers to the air-lubricated bearing of Kingsbury for experimental support of the theory.

The writer believes that the region where the author's results will be most useful lies at, or just above, the value of $\mu N/w$ which corresponds to the *minimum* coefficient of journal friction λ as depicted in Fig. 17. The reason why the theory may not apply very well *below* the minimum point has been shown by D. L.

Barnard, 4th, and co-authors in a contribution from the Massachusetts Institute of Technology presented before the American Chemical Society in 1923. Their recent experiments appear to prove that the unknown "oiliness" factors play an important part immediately below the minimum point.

On the other hand, for very *high* values of $\mu N/w$ corresponding to eccentricities less than, say, 20 per cent of the radial clearance, i.e., for $c < 0.2$, the author's theory, while correct, seems unnecessarily complex. It is sufficient in these ordinary high-speed problems to work with the simple formula of a concentric bearing, for example, Equation [**43**] of the writer's paper on Laws of Lubrication.[1]

At high speeds, moreover, it is difficult to predict the true viscosity of the oil film. The author's results here again are mathematically correct, but they presuppose a knowledge of that unknown viscosity. Suggestions for determining this coefficient of friction when thermal equilibrium has been reached, without knowing the true temperature or viscosity in the film, are to be found in the writer's paper, Lubrication Research Problems.[2]

A. E. Flowers. A word of caution should be given in regard to using Dr. McIlheny's tables for viscosities, which, though derived from the best data available at the time, were based on one or two models of Engler, Redwood, and Saybolt viscosimeters. Variations in different Saybolt viscosimeters may amount to 15 per cent, particularly for low viscosity. The American Society for Testing Materials is working in conjunction with the Bureau of Standards to standardize methods of viscosity measurements and to reduce the variations between readings on different Saybolt viscosimeters.

A factor of great importance in selecting the design for clearance and the viscosity of the oil is to see that the combination gives a margin of safety for both normal and any reasonable extreme of conditions that will still allow operation on the decreasing branch of the curve for the friction coefficient, rather than to select the conditions for minimum friction. With the minimum-friction condition for the usual operating conditions, any increase of load or increase of temperature (which always decreases the viscosity of oils) would increase the friction. This would still further increase the temperature and further decrease the viscosity. Accumulative effect, therefore, is a continuance of temperature rise and the tendency of the bearing to seize. Fortunately, seizing by this method inherently requires time. The length of time depends on the heat-absorbing capacity of the journals, bearings, supports, etc., whose temperature must be raised. This time gives an opportunity to observe the condition of overheating and to check it by extra cooling or the temporary use of a more viscous lubricant.

Bearing design, therefore, should provide not only a continuous

[1] Trans. A.S.M.E., vol. 37, p. 189.

[2] *Jour. Am. Soc. Naval Engrs.*, November, 1923.

heat-dissipating capacity, equal to the heat energy produced by friction, but also heat-absorbing capacity which will retard sudden temperature rises.

W. Elmer. It would be of service if the author could give us information on the practical application of his theories to particular problems. For instance, steam locomotives have now become so large, and the limit of sidewise clearance is so definite, that crankpin pressures have evidently reached a point exceeding the possibility of satisfactory lubrication. In a certain class of locomotives operating at 250 lb. per sq. in., the cylinder is $30\frac{1}{2}$ in. in diameter and the crankpin brasses are 9 by 9 in. The resulting bearing pressure is over 2000 lb. per sq. in. Under certain conditions of operation, for instance, on a grade, the pins will become overheated within two or three miles. If the speed is reduced, the pins may cool off, but the engine is then operating at a lower capacity than that for which it was designed.

This presents a real problem in railroad operation. Engines cannot be stopped every two or three miles to cool hot crankpins, nor is it feasible to so ease off that the engine will move the train without burning up the brasses. All kinds of lubrication have been tried, including hard grease, but the results are not what they should be. The author would be making a real contribution to railroad engineering if he could give us the information necessary to a solution of this problem.

W. E. Symons. The writer feels that the author has employed complex formulas to such an extent as to make his paper inapplicable to the practical problems that lubrication must necessarily solve. In the final analysis, if the results of a particular mathematical analysis or formula or the deductions and conclusions reached from that analysis are not applicable in a practical way to solve the problem of lubrication, the mathematical treatment is of little value.

From the standpoint of lubrication on railroads, there are today on more than 250,000 miles of railroad approximately 67,000 locomotives, 2,400,000 freight cars and about 56,000 passenger cars. Locomotive lubrication alone costs about $9,000,000 per year, and the locomotives have all the way from 14,000 to 77,000 sq. in. of lubricated area. The total amount spent for lubrication on railroads is probably at least $33,000,000 per year. The railroads are employing hundreds of high-grade practical engineers and an equal number with scientific training to solve the problems essential to the economical operation of trains. These men look to this Society to furnish them with definite and specific formulas and data, that they can follow and apply, to solve problems such as those of lubrication. If this paper will furnish a guide, it is a valuable contribution, but the writer fears that it is too scientific to be of much service for practical application.

It is to be hoped that the author in closing will supplement his paper with a clear, concise, specific statement, void of ambiguity and free from algebraic formulas, that any lubricating engineer of average intelligence can implicitly follow, as to: (*a*) Kind or characteristic and proper quantities of different lubricants; and (*b*) best methods, means, or devices to be used in their application, for railway, marine and industrial work or any specific section thereof, so that the lubricating engineer may know beyond the hazard of doubt that by following these instructions any given or specific problem can be satisfactorily handled.

James A. Hall. While railroad men are interested in lubricating problems involving high pressures and slow speeds, and probably conditions involving imperfect lubrication, in the machine-tool industry the problem of very light loads and high speeds is frequently encountered.

At the present time the writer is interested in the problem of improving grinding spindles, and making the bearings last somewhat longer than they now do. He has not gone very far, but hopes to receive some help from the present paper. However, he finds certain conditions which are not covered by the paper. For instance, a grinder with a 2-in. or 3-in. spindle, running at 1500 r.p.m., is off the lower end of the scale in Fig. 10 of clearance divided by radius or diameter. In regard to the lubricant used, even the thinnest lubricant would be so far up on the friction curve that it might reach infinity under certain conditions. It is a question whether or not we can expect, under the conditions given, with a clearance of, say, 0.001 or 0.0015 in. on a 2-in. or 3-in. journal, to maintain perfect lubrication throughout with the high speed and low pressure, and, if so, whether the coefficient of friction would not be far beyond the top of the chart.

In practice, with bearings with very small clearances it seems to be a fact that up to a certain point the bearing runs hotter as the oil supply is increased. For instance, a bearing seems to run, and by temperature measurements actually does run, cooler when lubricated at a rate of one drop per minute or one drop in two minutes than when, say, twenty drops per minute are fed into the bearing. This may possibly be due to the fact that the cooler oil coming in more rapidly might mean a lower viscosity.

W. T. Magruder. This paper is a good illustration of the different viewpoints of the man versed in mathematics and research and of the man who must make practical applications of the results of the research. The Society has had a number of papers which were beautiful from the theoretical point of view, and which pointed out the path for further experimental work, but which, when applied to practical problems, such as those cited in this discussion, failed to give the information desired for use by the practicing engineer.

The truly straight cylindrical bearing and uniform intensity of pressure do not exist except under zero load. A 3-in. shaft on 4-ft. centers, loaded so that it deflects 0.2 in., or a crankpin of a locomotive loaded so that the end has deflected 1/16 in., or more, from the position it occupied before loading, no longer have a straight line for the center of the shaft or pin. Instead, the center line is a continuous curve, and as shafting bearings do not support straight lines but curved lines, the problem is not simple, as the thickness of the oil film varies from a small fraction of its average thickness at the two ends where the intensity of pressure is the greatest to several times that thickness where there is no pressure. Mathematical analyses should take these practical facts into consideration, and so be of the greatest service.

S. Timoshenko[1] and J. M. Lessells. The first application of the hydrodynamic theory to an analytical solution of the problem of lubrication was made by N. P. Petroff[2] who developed the theory for the case of the central position of the journal in the bearing and found that the friction moment was independent of the load and proportional to the speed. Sommerfeld, to whom belongs the further development of the Reynolds theory, showed that at high speed his results coincided with the results of Petroff.

In 1913, W. J. Harrison[3] pointed out that the problem of the cylindrical bearing could be considerably simplified if the bearing was in the form of a complete cylinder surrounding the journal, and the entire space between the journal and its bearing filled with lubricant.

The graphical representation of the theoretical results obtained by the author is a very clear one, and this may encourage a general study of the theory of lubrication. So far, this theory, as at present developed, is not directly applicable to the design of bearings. It is limited by the assumptions made in deducing it. The most important of those is that relative to the continuity of the film. According to T. E. Stanton several attempts have been made at the National Physical Laboratory to set up Harrison's conditions in practice. Up to the present these attempts have failed for the reason that it has not yet been found possible to prevent rupture of the film on the lower half of the bush where the pressure was less than atmospheric, by an inflow of air from the ends of the bush.

The second point of importance is the effect of leakage of lubricant at the sides of the brass. In the case of thrust blocks the effect of this was worked out by A. G. M. Michell[4] with a square block having side leakage, and the permissible mean pressure was found to be 0.422 as much as when end leakage of the lubricant was

[1] Research Department, Westinghouse Electric and Manufacturing Company, East Pittsburgh, Pa.

[2] *Bulletin of the Institute of Technology*, St. Petersburg, 1885–1886.

[3] Trans. Cambridge Phil. Soc., vol. 28, 1913.

[4] *Zeitschrift für Mathematik und Physik*, vol. 52, 1905.

suppressed. Due to the limitations mentioned, the direct application of the theory to the design of bearings is difficult and the development of further experimental study of the problem becomes of great practical importance.

Factors such as heat flow and deflection of the shaft have to be considered. This, therefore, brings us to the consideration of tests on full-size bearings. Probably the most interesting work done along these lines is that of O. Lasche.[1] Confronted in his steam-turbine design with lubrication problems where the speed and the pressure were far beyond the usual practice, Lasche solved these problems in an experimental way by using full-size experimental bearings of 8-in. bore and 16-in. length. From these experiments some important conclusions were obtained on the distribution of pressure over the cylindrical surface of journal, the distribution of temperature, the effect of clearance, and the effect of different manner of conveying lubricant to the bearing. The value of further development of this kind of experiments in accordance with the new practical conditions cannot be overestimated.

Another field for experimental work represents the study of boundary lubrication in which the thickness of the film is of molecular dimensions. Some interesting results in this direction have been obtained recently by T. E. Stanton.[2]

The writers' opinion is that the theory, so far as at present developed, can give some guidance in proportioning bearings, but further development in its application will be possible only when the analytical study is accompanied by well-thought-out experimental work.

THE AUTHOR. Messrs. Barnard and Murphree have very kindly emphasized some important phases of the study of journal lubrication which the author did not mention, or purposely touched but lightly upon, being more intent upon a concise graphical presentation of W. J. Harrison's mathematical work. It is most fortunate that the discussion presented by all those who contributed to it also covers so many other important aspects of the subject.

The author's graphical presentation was limited to two dimensions in this first paper because it is the easiest direction of approach. It is also fairly adequate for practical needs. When the graphical study of partial bearings is completed the author feels that all variations from ideal bearings can be most profitably discussed.

When the statement is made that fact and theory disagree it is time to examine both the theory and the facts. If a simple theory is inadequate for practical purposes, a more comprehensive one must be brought forward. Obviously, the two-dimensional theory

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[1] Konstruction und Material im Bau von Dampfturbinen, by O. Lasche, Berlin, 1920.

[2] *Engineer*, vol. 135, June 29, 1923, p. 678.

of the complete bearing is valueless for a journal whose length is but a small fraction of its diameter, whether the speed be high or low. Obviously, also, it may be quite adequate for a journal three diameters long. It is also obvious that a simple graphical analysis of partial bearings would help to solve problems of incomplete lubrication of full bearings of any length.

The author does not believe that there is a sharp line of demarcation between complete fluid lubrication and the so-called "rupture" of the oil film. Careful observation should show a gradual passing from pure fluid friction to the friction of coated metal. If test conditions are such as to generate heat very rapidly the coatings would soon be destroyed and actual surface seizure might result.

Professor Marx calls attention to some student-thesis tests which are reported to have shown that for a full bearing the journal axis frequently runs above the horizontal plane through the bearing axis. The bearing to which he refers actually had two large oil-ring openings in the upper half which could reasonably be expected to prevent the realization of ideal pressure conditions. The unit pressures employed were very small and the method employed for charting the relative axis displacements would not show up such inconsistencies in the results as might actually exist. Consequently those theses should not be accepted as discrediting the theories developed by Reynolds and others.

Dr. Hersey calls attention to the "geometrical imperfections of bearing surfaces and freedom from distortion." These are of the greatest importance to investigators of the action of bearings in the region of film rupture. It becomes necessary for investigators to describe the mechanical methods employed to produce accurate bearing surfaces for their tests, so that some estimate of their probable exactness and dependability can be made by students of the test results. Here theory can assist by furnishing a standard by which to gage the influence of bearing-surface distortions and imperfections. The more perfect the test surfaces are, the closer they can be brought without interference between their high spots. The laws of viscosity will be found to hold for extremely thin films as demonstrated by Mr. Albert Kingsbury in 1900. He obtained film thicknesses of 1/40,000 in. without discovering any deviation from the law of absolute viscosity as accounting for bearing friction.

Dr. McIlheny's viscosity conversion tables to which Professor Flowers refers cover a Saybolt range from 67.8 to 500 sec. The author has used no values below 100 sec. Large conversion errors that might be expected for Saybolt viscosities around 50 sec. would not be present in the range used for plotting the curves in Fig. 10.

Reference to Fig. 16 will show how the minimum friction coefficient for a high-speed bearing can be obtained. At 1200 r.p.m. with a unit load of 60 lb. per sq. in. the lowest coefficient occurs

when the clearance is 0.005 in. per inch of diameter. This is probably fairly correct for a long journal. In practice, however, we find clearances much less than this. A customary clearance for steam-turbine bearings is 0.002 in. per inch of journal diameter. The friction in these bearings therefore appears to be much above the possible minimum for the pressure-speed conditions assumed. This minimum can be approached by increasing the customary clearance. It does not appear that such an increase of clearance would endanger the bearing, but quite the reverse.

A decrease of friction coefficient below the minimum for a given pressure-speed condition can be obtained by increasing the unit pressure. This at once gives a new pressure-speed condition. It would therefore be quite profitable to investigate the point raised by Professor Flowers to find how far down the decreasing branch of the friction curve of Fig. 10 it would be desirable to work in practice. This would amount to determining an allowable percentage of eccentricity, but it would have to be coupled with an allowable unit pressure.

Professor Hall raises some interesting points on grinder-spindle bearings using light loads, high speeds, and small clearances. For his purposes the chart section from $C = 0$ to $C = 0.1$ should be redrawn with very small subdivisions for C, and with additional curves for lower viscosity lubricants — not excluding one for air. Scales of smaller clearance would also be required. It is quite probable that the assumption of the existence of a complete oil film in a grinder-spindle bearing would have to be supplemented by the assumption that an extremely light oil is used. This suggestion is borne out by Professor Hall's statement that such bearings run hotter as the oil supply rate is increased, "up to a certain point." A constant rate of oil supply presupposes a constant rate of discharge. If the bearing design brings about a rapid discharge, it is to be expected that for a low supply rate the bearing lubrication would be imperfect so far as oil is concerned. It might even be running mainly on air, whose viscosity is very low, in which event the heating would be increased as the rate of oil supply is increased until a complete oil film was established. It is assumed that the rate of oil discharge is comparatively low and that most of the friction heat is disposed of by the bearing to the surrounding air.

The study of partial bearings, as previously stated, has not yet been completed by the author and he has therefore not yet made a direct comparison between the half-bearing characteristics determined graphically and analytically, the latter by Sommerfeld. The author has not yet studied the effect of friction, in a partial bearing, upon the direction of the resultant pressure. When this has been done his results should check exactly with those of Sommerfeld, and the small discrepancies pointed out by Professor Roberts should disappear.

Messrs. Timoshenko and Lessels have mentioned N. P. Petroff's

early development of the viscous-friction theory as applied to bearings when the axes of the journal and bearing coincide. This case gives a film of uniform thickness in which the friction differs but little from that of bearings with a small amount of eccentricity.

The hydrodynamic theory developed by Prof. Osborne Reynolds tied up the internal film pressures with the bearing speed and the viscosity of the lubricant. He proved that eccentricity was essential to the automatic generation of pressure within the film. Reynolds developed his theory first for a pair of inclined flat surfaces and then for a journal partially surrounded by its bearing, the latter corresponding with Tower's experimental bearing.

Sommerfeld solved the problem later for both the full bearing and half bearing, using polar coördinates. Harrison again solved the problem of the full bearing using rectilinear coördinates, the latter being directly comparable with the earlier work of Reynolds.

In all the above cases the end leakage was ignored because of the mathematical difficulties. The only mathematical study of end leakage to date was made in 1903 by A. G. M. Michell, for inclined flat surfaces only. His paper indicates that a prodigious amount of detail work was necessary to obtain his published results. It is to be hoped that some one with equal ability and perseverance will soon work out a simple mathematical analysis of journal-bearing lubrication, taking end leakage into account.

When such mathematical analyses as are at present available are presented graphically in a form that makes them readily available for checking experimental results, it will probably be found that simple factors can be determined that will answer all practical requirements for the designer and user of bearings. The author hopes that his work will prove a useful step in the desired direction.

Professor Magruder has called attention to the influence of journal deflection upon the mean thickness of the oil film. The author has made a preliminary study of this and will amplify the following statements in a later paper. The tendency in bearing design is toward shorter journals measured in terms of diameter. This reduces the danger from deflection. Self-aligning bearings are employed in some cases. Deflections are not very serious with short bearings using moderate pressures. Allowable deflections should be measured in terms of expected film thickness. When the author's study of film thicknesses of partial bearings is completed this question will be further discussed.

Messrs. Symons and Elmer have suggested that the author's work has not shed any very helpful light upon the every-day problems of railroad rolling-stock lubrication. The suggestion is appreciated as most timely. In the next section of his paper the author will try to bring the subject nearer to every-day problems by treating more fully the common cases of partial bearings.

It should then be possible to predict with fair accuracy what are

the most important characteristics in the lubrication of different combinations of bearing surfaces with a given journal.

The lubricant for bearings may be applied intermittently or continuously. Progress in bearing design must be made by making the lubrication as nearly continuous as possible. A reduction of the loss of oil from the bearing in intervals between application of oil will help much to bring about this condition. The difficulty experienced in lubricating crankpin bearings on locomotives is traceable to this weakness in their design.

No. 1907

THE BENDING AND TORSION OF MULTI-THROW CRANKSHAFTS ON MANY SUPPORTS

By S. Timoshenko,[1] East Pittsburgh, Pa.
Non-Member

In this paper the author considers the bending and torsion of multi-throw crankshafts on many supports. The assumptions made to simplify the problem are (1) *that the crankshaft is simply supported at the middle cross-sections of the journals, and* (2) *that each throw is considered as a rod whose cross-sections are small in comparison with the length. On these assumptions a system of equations for determination of bending moments at the supports is obtained. The application of these equations to the case of twist and bending of a three-throw crankshaft on four supports is considered in detail and the diagrams of bending moments for a numerical example are given.*

IN A PAPER entitled Torsion of Crankshafts,[2] which the author presented at the 1922 Annual Meeting of the Society, only the single-throw crankshaft was considered, and therefore the very considerable influence of the neighboring cranks on a throw was neglected. In this paper the equations for the torsion and bending of multi-throw crankshafts on many supports are given.

2 In applying these equations to the case of twist of a three-throw crankshaft, it is found that the effects of constraint on the "reduced length" of the first and third throws are small (about 3 per cent). The same effect for the second throw is about half that obtained in the above-mentioned paper on the assumption of journals clamped at the middle cross-sections. The bending of the three-throw crankshaft is also considered and the bending-moment diagrams are given for a numerical example.

3 The assumptions made to simplify the problem are (*a*) that the crankshaft is simply supported at the middle cross-sections of the journals and (*b*) that each throw is considered as a bar *mnpqst*, Fig. 1, whose cross-sections are small in comparison with the lengths r, e, and f. It will be shown that the calculations made on

[1] Research Dept., Westinghouse Elec. & Mfg. Co.
[2] Trans. A.S.M.E., vol. 44, p. 653.

Presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

this latter assumption are in good accord with the results obtained in the author's previous paper on the basis of a more detailed consideration of the deformation of a single throw.

Definitions and Symbols

4 The definitions used are those employed in the author's previous paper. The dimensions are shown in Fig. 1. The principal symbols are as follows:

C = torsional rigidity
B = flexural rigidity
G = modulus of shear
E = Young's modulus

$C_1 = G\theta_1 = \frac{\pi}{32} d_1^4 G$ = torsional rigidity of the journal and the crankpin

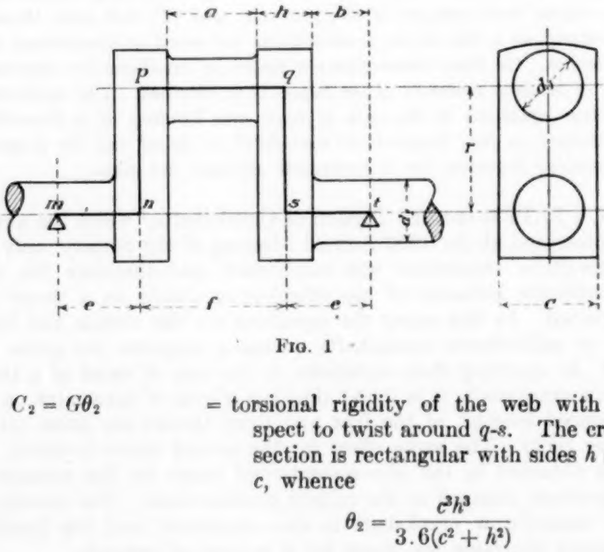


Fig. 1

$C_2 = G\theta_2$ = torsional rigidity of the web with respect to twist around q-s. The cross-section is rectangular with sides h and c, whence

$$\theta_2 = \frac{c^3h^3}{3.6(c^2 + h^2)}$$

$B_1 = EI_1 = \frac{\pi}{64} d_1^4 E$ = flexural rigidity of the journal and the crankpin

$B_2 = EI_2 = \frac{hc^3}{12} E$ = flexural rigidity of the web against bending in the plane through q-s perpendicular to the plane of the drawing of Fig. 1.

$B_3 = EI_3 = \frac{ch^3}{12} E$ = flexural rigidity of the web against bending in the plane of the throw.

5 The bending and torsion of the multi-throw crankshaft are calculated by the same method as for a continuous beam. Analysis is first made of the deformations of a single crank due to various forces and moments. The bending moments at the supports are then determined by using the conditions of continuity of the elastic line. The axes of coördinates for each throw are chosen as shown in Fig. 2. The xy-plane always coincides with the plane of the throw and the z-axis is so directed that rotation of a right-hand screw from y to z would produce motion in the positive direction of the x-axis.

6 The external forces on the throw to be considered are:

a The force of the connecting rod on the crankpin. This, acting in a plane perpendicular to the x-axis, is resolved

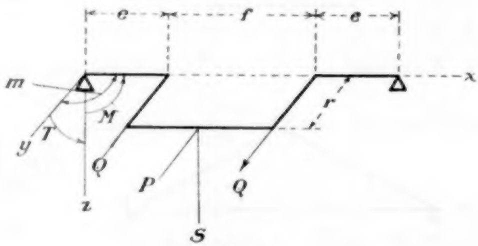


Fig. 2

into two components, P and S, parallel to the y- and z-axes, respectively (Fig. 2)

b The centrifugal forces due to the web and crankpin. These are represented by two equal forces, Q, parallel to the y-axis (Fig. 2)

c The action of the neighboring cranks at the middle cross-section of the journal. This action is represented by a shearing force and a couple which usually is resolved into three components, namely,

m, the bending moment in the plane of the throw

M, the bending moment in the plane perpendicular to the plane of the throw, and

T, the twisting moment.

The effects due to these different groups of forces will be analyzed separately and the combined effect obtained by the method of superposition of the effects of forces.

Bending in the Plane of the Throw

7 Referring to Fig. 3(a), consider the bending of the throw produced by the forces P, Q, and by a couple m acting at the left support in the xy-plane. The relation between the positive sense

of the moment m and the positive direction of the z-axis is the same as the relation of rotation to translation in a right-hand screw. The same rule is used in determining the sign of the angle of rotation of any cross-section of the journal. In accordance with this, the angle φ_1 of rotation of the left terminal cross-section is positive, and φ_2 negative.

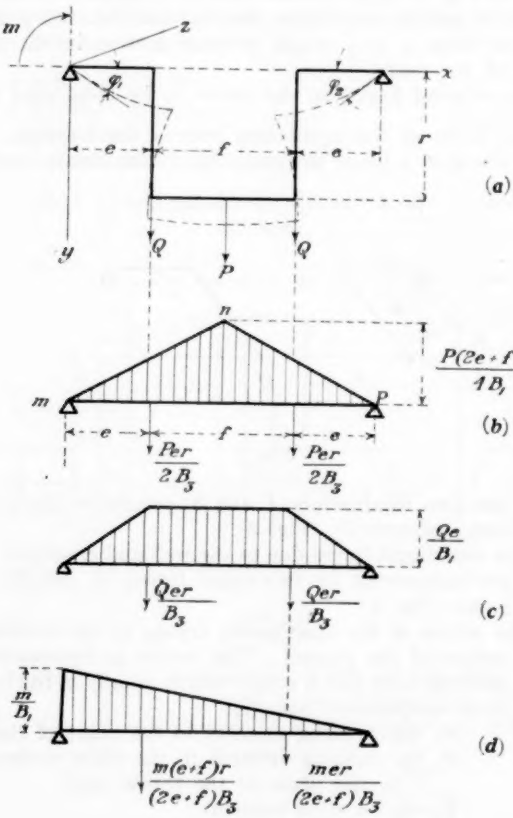


FIG. 3

8 In calculating these angles, Mohr's method[1] is the most expeditious. In the case of a simple beam, the assumption must be made that it sustains a continuous distributed load the value of whose intensity at each cross-section is numerically equal to M/B, where M denotes the bending moment at the given cross-section and B the flexural rigidity of the section. The reactions

[1] Abhandlungen aus dem Gebiete der Technischen Mechanik, by O. Mohr.

at the supports calculated for this imaginary loading give the numerical value of the angles of rotation of the terminal cross-section of the beam. In the case of the crankshaft this imaginary loading gives the angular changes due to the bending of the journals and crankpin. In order to take account of the deformation due to the webs, two additional concentrated forces [see (*a*) and (*b*), Fig. 3] numerically equal to the angular deformation of the webs must be added.

9 The angles φ_1, φ_2, of rotation at the supports of the throw will be calculated in their component parts (φ_1', φ_2'), $(\varphi_1'', \varphi_2'')$, $(\varphi_1''', \varphi_2''')$ due respectively to the forces P, Q, and the moment m.

10 In the case of bending by the force P, the distributed load is represented by the triangle *mnp*, Fig. 3(*b*), and the additional concentrated load at each web is $\frac{Per}{2B_3}$ [Fig. 3(*b*)]. This quantity is readily obtained. Since the webs are each uniformly bent into circular arcs by the couples $\frac{Pe}{2}$, the curvature, $\frac{1}{\rho}$, from the formula $\frac{1}{\rho} = \frac{M}{EI}$, is equal to $\frac{Pe}{2B_3}$, and the angular deformation is $\frac{Per}{2B_3}$. Then

$$\varphi_1' = -\varphi_2' = P\left[\frac{(2e+f)^2}{16B_1} + \frac{er}{2B_3}\right]$$

The centrifugal forces Q give, by the same rule, the continuous loading and concentrated forces as shown in Fig. 3(*c*). Hence

$$\varphi_1'' = -\varphi_2'' = Q\left[\frac{e(e+f)}{2B_1} + \frac{er}{B_3}\right]$$

Similarly, the moment m, Fig. 3(*d*), gives

$$\varphi_1''' = m\left[\frac{2e+f}{3B_1} + \frac{e^2r}{(2e+f)^2B_3} + \frac{(e+f)^2r}{(2e+f)^2B_3}\right]$$

$$\varphi_2''' = -m\left[\frac{2e+f}{6B_1} + \frac{2e(e+f)r}{(2e+f)^2B_3}\right]$$

The sum of the effects of all the forces shown in Fig. 3(*a*), then, is

$$\varphi_1 = m\alpha_1 + Pt_1 + Qt_2 \quad \ldots\ldots\ldots\ldots \quad [1]$$

$$\varphi_2 = -m\alpha_2 - Pt_1 - Qt_2 \quad \ldots\ldots\ldots\ldots \quad [2]$$

where

$$\alpha_1 = \frac{2e+f}{3B_1} + \frac{e^2r}{(2e+f)^2B_3} + \frac{(e+f)^2r}{(2e+f)^2B_3} \quad \ldots\ldots \quad [3]$$

$$\alpha_2 = \frac{2e+f}{6B_1} + \frac{2e(e+f)r}{(2e+f)^2B_3} \quad \ldots\ldots\ldots\ldots \quad [4]$$

$$t_1 = \frac{(2e+f)^2}{16B_1} + \frac{er}{2B_3} \quad \text{[5]}$$

$$t_2 = \frac{e(e+f)}{2B_1} + \frac{er}{B_3} \quad \text{[6]}$$

Bending in the Plane Perpendicular to the Plane of the Throw

11 The bending of the throw by the force S, parallel to the z-axis, and a couple M, acting in the xz-plane, Fig. 4(a), is now to

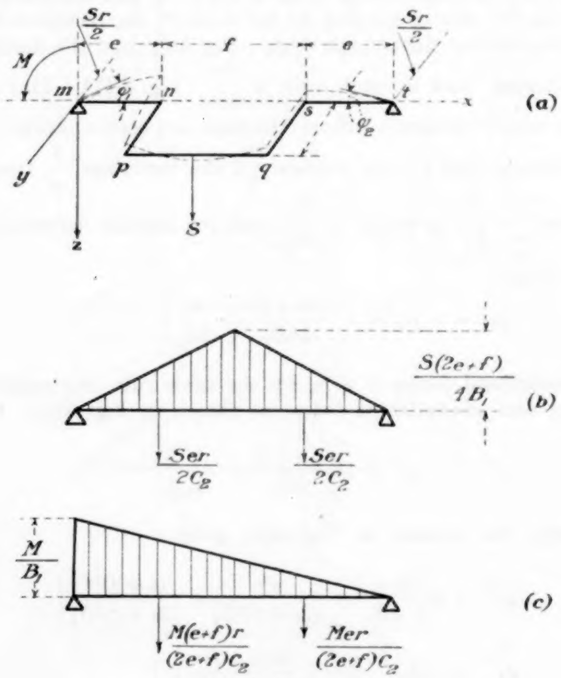


Fig. 4

be considered. A moment $Sr/2$ about the x-axis is applied at each support to prevent rotation under the action of the force S.

12 By the same method as above, the angles of rotation produced by the force S [Mohr loading given in Fig. 4(b)] are

$$\psi_1 = -\psi_2 = -S\left[\frac{(2e+f)^2}{16B_1} + \frac{er}{2C_2}\right] \quad \text{[1]}$$

13 The case of M is more complicated. The angles of rotation of the ends are affected not only by the angular deformation of different parts of the throw in the plane of M, but also by the

deflection of the webs and the twist of the crankpin. The former can be calculated by the Mohr method as above [loading as in Fig. 4(c)]. These angles are

$$\left.\begin{aligned} \psi_1'' &= M\left[\frac{2e+f}{3B_1} + \frac{e^2r}{(2e+f)^2C_2} + \frac{(e+f)^2r}{(2e+f)^2C_2}\right] \\ \psi_2'' &= -M\left[\frac{2e+f}{6B_1} + \frac{2e(e+f)r}{(2e+f)^2C_2}\right] \end{aligned}\right\} \quad . \qquad [\mathrm{II}]$$

14 The rotation of the ends due to the bending of the webs and to the twist of the crankpin can be taken from Fig. 5(a), where the corresponding distortion of the crankshaft is shown. It is

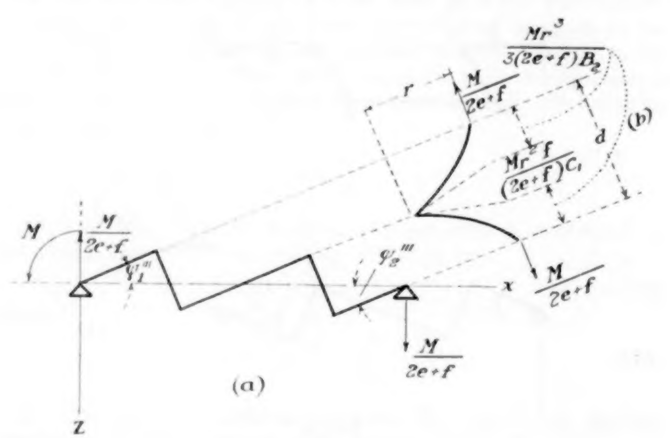


FIG. 5

seen that these angles are each equal to $d/(2e+f)$, where d is the displacement shown in Fig. 5(b). This displacement consists of twice the deflection of the web and the displacement due to the twist of the crankpin. The deflection of the web considered as a cantilever is $\frac{M}{2e+f}\,\frac{r^3}{3B_2}\cdot$ The displacement due to the twist is obtained by multiplying the angle of twist by the length r and equals $\frac{Mr}{2e+f}\,\frac{rf}{C_1}\cdot$ In this way

$$d = M\left[\frac{2r^3}{3(2e+f)B_2} + \frac{fr^2}{(2e+f)C_1}\right]$$

and

$$\psi_1''' = \psi_2''' = \frac{d}{2e+f} = M\left[\frac{2r^3}{3(2e+f)^2B_2} + \frac{fr^2}{(2e+f)^2C_1}\right] \quad . \qquad [\mathrm{III}]$$

15 The complete angles of rotation, equal to the sum of [I], [II], and [III], are:

$$\psi_1 = \beta_1 M - uS \qquad [7]$$

$$\psi_2 = -\beta_2 M + uS \qquad [8]$$

where

$$\beta_1 = \frac{2e+f}{3B_1} + \frac{e^2 r}{(2e+f)^2 C_2} + \frac{(e+f)^2 r}{(2e+f)^2 C_2} + \frac{2r^3}{3(2e+f)^2 B_2} + \frac{fr^2}{(2e+f)^2 C_1} \qquad [9]$$

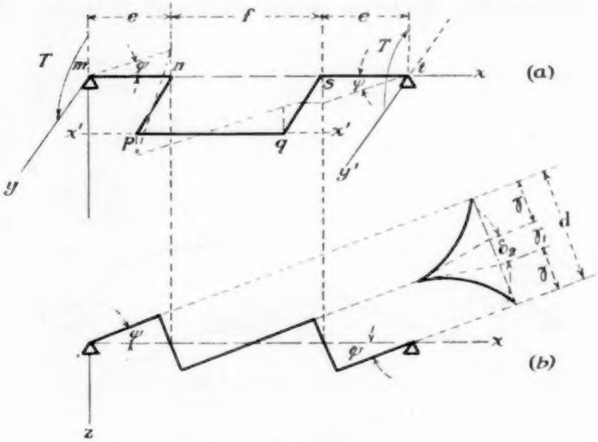


Fig. 6

$$\beta_2 = \frac{2e+f}{6B_1} + \frac{2e(e+f)r}{(2e+f)^2 C_2} - \frac{2r^3}{3(2e+f)^2 B_2} - \frac{fr^2}{(2e+f)^2 C_1} \qquad [10]$$

$$u = \frac{(2e+f)^2}{16B_1} + \frac{er}{2C_2} \qquad [11]$$

16 These, then, are the angles of rotation of the terminal cross-sections about axes parallel to the y-axis, produced by the forces and moments given in Fig. 4(a). As to rotation about the z-axis produced by these forces, conclusions can at once be drawn from a comparison of the cases represented in Figs. 3 and 4 and the use of the well-known reciprocal theorem. It is easy to see that the forces of Fig. 3, i.e., those in the plane xy of the throw, produce displacements in the xy-plane only. Reciprocally, then, the forces in the xz-plane will not produce displacements in the xy-plane and therefore there will be no rotation produced by them about the z-axis. The rotation about the x-axis produced by the forces of Fig. 4(a) will be considered below.

Twist of Simply Supported Throw

17 Fig. 6 is a diagrammatic representation of a throw twisted by the moments T applied at the middle cross-sections of the journals. As before, the dotted lines represent the deformed crankshaft. The total twist consists of the sum of the deformations of the portions e of the journals, of the crankpin, and of the two webs. Now if δ_1 and δ_2 are the angles of twist of the journal and crankpin, respectively,

$$\delta_1 = \frac{Te}{C_1}; \quad \delta_2 = \frac{Tf}{C_1}$$

18 Further, the bending of each web produces the angular displacement δ_3, equal to the angle between the tangents to the curve of flexure at its ends. The bending moment is equal to the twisting moment T. Since the moment at each cross-section is constant, the curve of flexure of the web is a circle. Considering it as a beam of length r,

$$\delta_3 = \frac{Tr}{B_2}$$

19 The total relative angular displacement, δ, measured at the middle sections of the journals, will then be

$$\delta = 2\delta_1 + \delta_2 + 2\delta_3 = Tw \quad \ldots\ldots \quad [12]$$

where

$$w = \frac{2e+f}{C_1} + \frac{2r}{B_2} \quad \ldots\ldots\ldots\ldots \quad [12a]$$

20 The couples T produce not only the twist δ but also the rotation ψ of the terminal cross-sections about the y- and y'-axes. This rotation, due to the bending of webs and the twist of the crankpin [Fig. 6(b)], is calculated as in Par. 14. Then

$$\psi = \frac{d}{2e+f}$$

The flexure of the web produces a deflection

$$\gamma = \frac{Tr^2}{2B_2}$$

and the displacement due to the twist of the crankpin is

$$\gamma_1 = \delta_2 r = \frac{Tfr}{C_1}$$

Then

$$d = 2\gamma + \gamma_1 = T\left(\frac{r^2}{B_2} + \frac{fr}{C_1}\right)$$

and

$$\psi = Ts \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [13]$$

where

$$s = \frac{1}{2e + f}\left(\frac{r^2}{B_2} + \frac{fr}{C_1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [14]$$

21 Now the twist of the throw produced by the forces of Fig. 4(*a*) can be easily found; denoting this twist by δ_0, and applying the reciprocal theorem to the cases of Figs. 4(*a*) and 6(*a*), it is found that

$$M\psi = T\delta_0$$

or, employing [**13**] and [**14**],

$$\delta_0 = \frac{M\psi}{T} = Ms \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

This twist is in the direction of the twisting couple T of Fig. 6(*a*). When there are two bending moments, M_1 and M_2, one at each support, the corresponding twist is obtained in the same manner and will be equal to

$$\delta_0 = (M_1 + M_2)s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15a]$$

22 If, in the case of Fig. 4 (*a*), M be made equal to zero and in Fig. 6(*a*) the moment $T + (Sr/2)$ be substituted for T, the combination of these two cases gives that shown in Fig. 7. In the further development of the subject the values of the angles ψ_1 and ψ_2, Fig. 7, are necessary. Using Equations [**5**], [**8**], and [**13**],

$$\psi_1 = \left(T + \frac{Sr}{2}\right)s - uS$$

$$\psi_2 = \left(T + \frac{Sr}{2}\right)s + uS$$

By using [**11**] and [**14**] these equations can be written as follows:

$$\psi_1 = Ts - Sv_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [16]$$

$$\psi_2 = Ts + Sv_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where

$$v_1 = u - \frac{rs}{2} = \frac{(2e + f)^2}{16B_1} + \frac{er}{2C_2} - \frac{r}{2(2e + f)}\left(\frac{r^2}{B_2} + \frac{fr}{C_1}\right) \quad . \quad . \quad . \quad [18]$$

$$v_2 = u + \frac{rs}{2} = \frac{(2e + f)^2}{16B_1} + \frac{er}{2C_2} + \frac{r}{2(2e + f)}\left(\frac{r^2}{B_2} + \frac{fr}{C_1}\right) \quad . \quad . \quad . \quad [19]$$

Twist of Throw Clamped at Middle Cross-section of Journals

23 The complete solution of this problem was given in the author's previous paper [1] (Case III, Partial Constraint) where the deformations of all parts of the throw were taken into consideration. These results will now be compared with those obtained by the

[1] Torsion of Crankshafts, Trans. A.S.M.E., vol. 44, p. 653.

simplified methods of this paper in order to obtain some idea of their exactness.

24 The solution of the problem will be found by superimposing the solution of the case shown in Fig. 8 on that shown in Fig. 6(*a*). The couples M of Fig. 8 must be chosen in such a manner as to annihilate the angles of rotation ψ produced by the twisting moments T, Fig. 6(*a*). In this way we obtain the equation

$$\psi = \psi' \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathrm{IV}]$$

The angle ψ is given by Equations [**13**] and [**14**]. The angle ψ' is calculated by using Equations [**7**] and [**8**] of the case shown in Fig. 4(*a*). It is only necessary to put $S = 0$ in these equations and

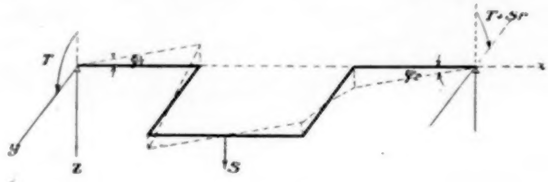


FIG. 7

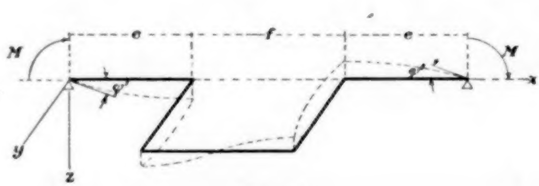


FIG. 8

to take into account the fact that in Fig. 8 there are two terminal couples M acting in the same direction. Then

$$\psi' = (\beta_1 - \beta_2)M \quad . \; . \; . \; . \; . \; . \; . \; . \quad [\mathrm{V}]$$

where β_1 and β_2 are given by Equations [**9**] and [**10**]. Substituting [V] and [**13**] in Equation [IV],

$$M = T\frac{s}{\beta_1 - \beta_2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{20}]$$

The angle of twist δ_1 for this case will be

$$\delta_1 = \delta - 2\delta_0 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathrm{VI}]$$

where δ is the twist of the throw simply supported, Equation [**12**], and δ_0, given by Equation [**15**], is the decrease of twist due to the bending moment M. The quotient δ_1/δ gives the ratio in which the reduced length of the throw decreases as a result of constraint.

Substituting Equation [**20**] in [**15**] and using [**12**], there will be obtained from [VI]

$$\frac{\delta_1}{\delta} = 1 - \frac{2\delta_0}{\delta} = 1 - \frac{2s^2}{(\beta_1 - \beta_2)w} \quad . \; . \; . \; . \; . \quad [\mathbf{21}]$$

where β_1, β_2, s, and w are given by Equations [**9**], [**10**], [**14**] and [**12a**].

Numerical Example

25 Referring to Fig. 1, let $d_1 = 10.25$ in., $a = 13$ in., $r = 11$ in., $b = 6.5$ in., $h = 5.5$ in., $c = 14$ in., whence $e = 9.25$ in. and $f = 18.5$ in.

$$B_1 = \frac{\pi \times 10.25^4 E}{64} = 542E; \quad B_2 = \frac{5.5 \times 14^3 E}{12} = 1258E$$

$$C_1 = \frac{\pi \times 10.25^4 G}{32} = 1085G; \quad C_2 = \frac{14^3 \times 5.5^3 G}{3.6(14^2 + 5.5^2)} = 562G$$

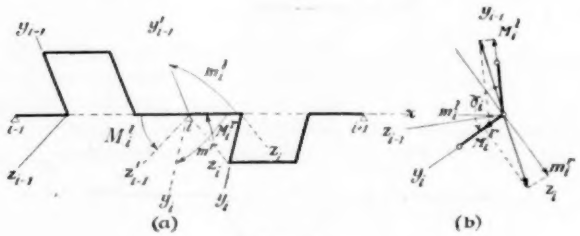


Fig. 9

Taking $E/G = 2.6$ and substituting this value in [**21**],

$$\frac{\delta_1}{\delta} = 1 - \frac{0.0058}{0.0408} = 1 - 0.142.$$

Therefore the reduced length, as compared with the case of no constraint, is decreased by 14.2 per cent. The detailed calculations made for the same numerical example in the author's previous paper gave a reduction of 12.5 per cent. This shows that the simplified equations give a good approximation, and they will be used further in the consideration of multi-throw crankshafts.

The Multi-Throw Crankshaft

26 Consider now the crankshaft with many throws. Assume it to be simply supported at the middle cross-sections of the journals and number these supports in order 1, 2, 3 . . . . Then the throw lying between supports $i - 1$ and i will be the $i - 1$th throw, and all forces acting on it will have this subscript. When the twisting and bending moments at every support are known it is easy to calculate the deformations of any throw from the

foregoing equations. The twisting moment at any support can be calculated without difficulty from the equations of statics. For calculating the bending moments, the condition of continuity of the elastic curve at the supports must be taken into consideration. In Fig. 9 two consecutive throws cut out of a multi-throw crankshaft and having supports $i-1$, i, and $i+1$, are represented. The bending moment at the support i is resolved into two components, namely, the components m^l_i and M^l_i when the throw on the left of the support i is considered; and into the components m^r_i and M^r_i for the throw on the right of the support i.

27 As before, m denotes the bending moment in the plane of the throw considered and M the bending moment in the plane perpendicular to the plane of the throw. The positive directions of these moments are shown in Fig. 9(a). The representation of the same moments by vectors is given in Fig. 9(b). As m^l_i and M^l_i on the one side, and m^r_i and M^r_i on the other represent the same bending moment, the resultant of m^l_i and M^l_i must be

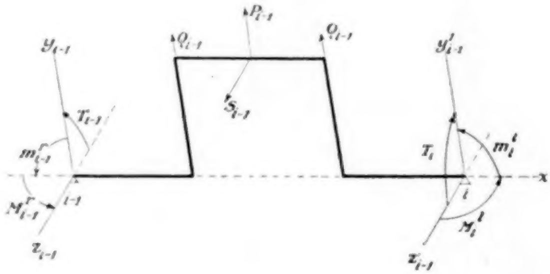


FIG. 10

equal and opposite to the resultant of m^r_i and M^r_i as shown in Fig. 9(b). Therefore, denoting by γ_i the angle between the consecutive throws at i and resolving the couples in the directions y_{i-1} and z_{i-1},

$$\left.\begin{aligned} M^l_i + M^r_i \cos\gamma_i - m^r_i \sin\gamma_i = 0 \\ m^l_i + M^r_i \sin\gamma_i + m^r_i \cos\gamma_i = 0 \end{aligned}\right\} \quad . \ . \ . \ . \quad [22]$$

28 Two more equations are obtained from the conditions of continuity. Consider the throw on the left of the support i. The positive directions of all the forces and couples acting on this throw are shown in Fig. 10. Denote by φ^l_i and ψ^l_i the angles of rotation of the cross-section at the support i about the axes z'_{i-1} and y'_{i-1}, respectively. The same rule holds for the sign of the angles as before.

29 Using Equations [1] and [2] and taking into account the directions of the couples m^r_{i-1} and m^l_i and of the forces Q_{i-1} and P_{i-1},

$$\varphi^l_i = m^l_i\alpha_1 - m^r_{i-1}\alpha_2 - P_{i-1}t_1 - Q_{i-1}t_2 \quad . \ . \ . \quad [23]$$

where α_1, α_2, t_1, and t_2 are given by Equations [**3**], [**4**], [**5**], and [**6**]. In the same manner, by using Equations [**7**], [**8**], [**16**], and [**17**],

$$\psi^l_i = M^l_i\beta_1 - M^r_{i-1}\beta_2 + T_{i-1}s + S_{i-1}v_2 \quad . \; . \; . \quad [\mathbf{24}]$$

where β_1, β_2, s, and v_2 are given by Equations [**9**], [**10**], [**14**], and [**19**].

30 Considering now the throw on the right of the support i and denoting by φ^r_i and ψ^r_i the angles of rotation of the cross-section at i about the z_i- and y_i-axes of this throw, the following equations are obtained in the same manner as above:

$$\left.\begin{aligned} \varphi^r_i &= m^r_i\alpha_1 - m^l_{i+1}\alpha_2 + P_i t_1 + Q_i t_2 \\ \psi^r_i &= M^r_i\beta_1 - M^l_{i+1}\beta_2 + T_i s - S_i v_1 \end{aligned}\right\} \quad . \; . \; . \; . \quad [\mathbf{25}]$$

Here $T_i = T_{i-1} + S_{i-1}r$ represents the twisting moment at the support i, and the coefficient v_1 is given by Equation [**18**]. The

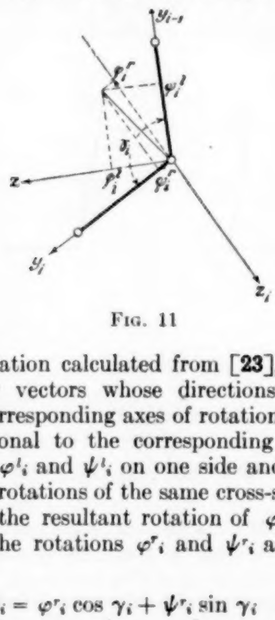


Fig. 11

small angles of rotation calculated from [**23**], [**24**], and [**25**] can be represented by vectors whose directions coincide with the directions of the corresponding axes of rotation [1] and whose magnitudes are proportional to the corresponding angles of rotation, Fig. 11. Now as φ^l_i and ψ^l_i on one side and φ^r_i and ψ^r_i on the other represent the rotations of the same cross-section of the journal at the support i, the resultant rotation of φ^r_i and ψ^l_i must be equal to that of the rotations φ^r_i and ψ^r_i as shown in Fig. 11. From this figure

$$\left.\begin{aligned} \varphi^l_i &= \varphi^r_i \cos\gamma_i + \psi^r_i \sin\gamma_i \\ \psi^l_i &= -\varphi^r_i \sin\gamma_i + \psi^r_i \cos\gamma_i \end{aligned}\right\} \quad . \; . \; . \; . \quad [\mathbf{26}]$$

31 These equations together with those of [**22**] give a system of four equations for the support i. Analogous equations can be written for all the intermediate supports. In this manner are obtained a number of equations equal to the number of unknown

[1] There exists between the positive direction of the axis of rotation and the direction of rotation the same relation as between rotation and translation in a right-handed screw.

quantities m^l_i, m^r_i, M^l_i, M^r_i, where $i = 1, 2, 3 \ldots . n$, and n is the number of supports.

32 When Equations [**23**], [**24**], and [**25**] are substituted in [**26**], those of the latter number take the following form:

$$\left.\begin{aligned} & m^l_i\alpha_1 - m^r_{i-1}\alpha_2 - P_{i-1}t_1 - Q_{i-1}t_2 = (m^r_i\alpha_1 - m^l_{i+1}\alpha_2 + P_it_1 \\ & \quad + Q_it_2)\cos\gamma_i + (M^r_i\beta_1 - M^l_{i+1}\beta_2 - S_iv_1 + T_is)\sin\gamma_i \\ & M^l_i\beta_1 - M^r_{i-1}\beta_2 + T_{i-1}s + S_{i-1}v_2 = (M^r_i\beta_1 - M^l_{i+1}\beta_2 \\ & \quad - S_iv_1 + T_is)\cos\gamma_i - (m^r_i\alpha_1 - m^l_{i+1}\alpha_2 + P_it_1 \\ & \quad + Q_it_2)\sin\gamma_i \end{aligned}\right\} \quad [\mathbf{27}]$$

Now equations of the type of [**22**] give

$$\left.\begin{aligned} & m^l_i = -\frac{M^r_i + M^l_i\cos\gamma_i}{\sin\gamma_i}; \qquad m^r_i = \frac{M^l_i + M^r_i\cos\gamma_i}{\sin\gamma_i} \\ & m^r_{i-1} = \frac{M^l_{i-1} + M^r_{i-1}\cos\gamma_{i-1}}{\sin\gamma_{i-1}}; \; m^l_{i+1} = -\frac{M^r_{i+1} + M^l_{i+1}\cos\gamma_{i+1}}{\sin\gamma_{i+1}} \end{aligned}\right\} [\mathbf{28}$$

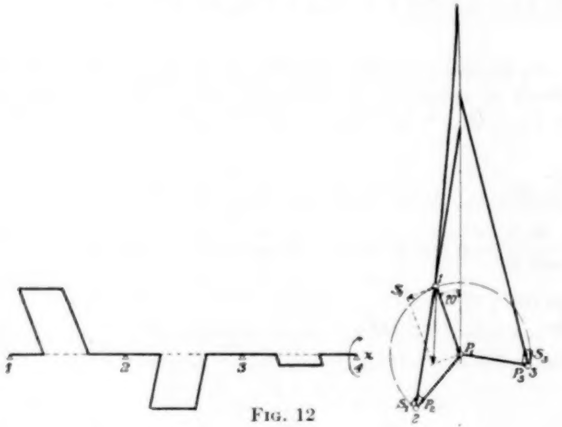


Fig. 12

Substituting these values in [**27**],

$$\left.\begin{aligned} & -\frac{\alpha_2}{\sin\gamma_{i-1}}M^l_{i-1} - \alpha_2\cot\gamma_{i-1}M^r_{i-1} - 2\alpha_1\cot\gamma_iM^l_i \\ & \quad - \left(\frac{\alpha_1}{\sin\gamma_i} + \frac{\alpha_1\cos^2\gamma_i}{\sin\gamma_i} + \beta_1\sin\gamma_i\right)M^r_i + \left(\beta_2\sin\gamma_i\right. \\ & \quad \left. - \alpha_2\frac{\cos\gamma_i\cos\gamma_{i+1}}{\sin\gamma_{i+1}}\right)M^l_{i+1} - \alpha_2\frac{\cos\gamma_i}{\sin\gamma_{i+1}}M^r_{i+1} = P_{i-1}t_1 \\ & \quad + Q_{i-1}t_2 + (P_it_1 + Q_it_2)\cos\gamma_i - S_iv_1\sin\gamma_i + T_is\sin\gamma_i. \\ & -M^r_{i-1}\beta_2 + (\alpha_1 + \beta_1)M^l_i + (\alpha_1 - \beta_1)\cos\gamma_iM^r_i \\ & \quad + \left(\beta_2\cos\gamma_i + \alpha_2\frac{\sin\gamma_i\cos\gamma_{i+1}}{\sin\gamma_{i+1}}\right)M^l_{i+1} + \alpha_2\frac{\sin\gamma_i}{\sin\gamma_{i+1}}M^r_{i+1} \\ & \quad = -T_{i-1}s + T_is\cos\gamma_i - S_{i-1}v_2 - S_iv_1\cos\gamma_i - (P_it_1 \\ & \quad + Q_it_2)\sin\gamma_i \end{aligned}\right\} \quad [\mathbf{29}]$$

These equations play the same rôle in calculating multi-throw crankshafts as the well-known equations of Clapeyron for the calculation of continuous beams.

33 *Special Cases of Equations* [**29**]. (*a*) When $\gamma_2 = \gamma_3 = \gamma$, Equations [**29**] take the following simplified form:

$$\left.\begin{aligned}
&-\frac{\alpha_2}{\sin\gamma} M^l_{i-1} - \alpha_2 \cot\gamma M^r_{i-1} - 2\alpha_1 \cot\gamma M^l_i \\
&\quad -\left(\frac{\alpha_1}{\sin\gamma} + \alpha_1\cos\gamma\cot\gamma + \beta_1\sin\gamma\right) M^r_i + (\beta_2\sin\gamma \\
&\quad - \alpha_2\cos\gamma\cot\gamma) M^l_{i+1} - \alpha_2\cot\gamma M^r_{i+1} = P_{i-1}t_1 + Q_{i-1}t_2 \\
&\quad + (P_i t_1 + Q_i t_2)\cos\gamma - S_i v_1 \sin\gamma + T_i s \sin\gamma. \\
&- M^r_{i-1}\beta_2 + (\alpha_1 + \beta_1) M^l_i + (\alpha_1 - \beta_1)\cos\gamma\, M^r_i \\
&\quad + (\beta_2 + \alpha_2)\cos\gamma\, M^l_{i+1} + \alpha_2 M^r_{i+1} = -T_{i-1}s + T_i s\cos\gamma \\
&\quad - S_{i-1}v_2 - S_i v_1 \cos\gamma - (P_i t_1 + Q_i t_2)\sin\gamma
\end{aligned}\right\} [\mathbf{30}]$$

34 (*b*) Equations [**30**] take the form given below when the crankshaft is subjected to twisting only. Substituting therein $P_{i-1} = P_i = Q_{i-1} = Q_i = S_{i-1} = S_i = 0$ and putting $T_{i-1} = T_i = T$,

$$\left.\begin{aligned}
&-\frac{\alpha_2}{\sin\gamma} M^l_{i-1} - \alpha_2 \cot\gamma\, M^r_{i-1} - 2\alpha_1 \cot\gamma\, M^l_i \\
&\quad -\left(\frac{\alpha_1}{\sin\gamma} + \alpha_1\cos\gamma\cot\gamma + \beta_1\sin\gamma\right) M^r_i + (\beta_2\sin\gamma \\
&\quad - \alpha_2\cos\gamma\cot\gamma) M^l_{i+1} - \alpha_2\cot\gamma\, M^r_{i+1} = Ts\sin\gamma. \\
&- \beta_2 M^r_{i-1} + (\alpha_1 + \beta_1) M^l_i + (\alpha_1 - \beta_1)\cos\gamma\, M^r_i \\
&\quad + (\alpha_2 + \beta_2)\cos\gamma\, M^l_{i+1} + \alpha_2 M^r_{i+1} = Ts(\cos\gamma - 1)
\end{aligned}\right\} [\mathbf{31}]$$

Application of Equations to the Three-Throw Crankshaft

35 In the case of the three-throw crankshaft,

$$\gamma_2 = \gamma_3 = \gamma = 120°,\ \sin\gamma = \tfrac{1}{2}\sqrt{3},\ \cos\gamma = -\tfrac{1}{2},\ \cot\gamma = -\frac{1}{\sqrt{3}}.$$

Since the shaft is assumed to be simply supported, the bending moments at the terminal cross-sections are equal to zero, i.e.,

$$M^l_1 = M^r_1 = M^l_4 = M^r_4 = 0$$

36 *General Formula for Three-Throw Shaft.* Under the assumptions of the preceding paragraph Equations [**30**] for the two intermediate supports become

$$\left.\begin{aligned}
&(\alpha_1+\beta_1)M^l_2-\tfrac{1}{2}(\alpha_1-\beta_1)M^r_2-\tfrac{1}{2}(\alpha_2+\beta_2)M^l_3+\alpha_2M^r_3\\
&\quad=-T_1s-\tfrac{1}{2}T_2s-S_1v_2+\tfrac{1}{2}S_2v_1-\frac{\sqrt{3}}{2}(P_2t_1+Q_2t_2)\\
&\alpha_1M^l_2-(\tfrac{5}{4}\alpha_1+\tfrac{3}{4}\beta_1)M^r_2+(\tfrac{3}{4}\beta_2-\tfrac{1}{4}\alpha_2)M^l_3+\tfrac{1}{2}\alpha_2M^r_3\\
&\quad=\tfrac{3}{4}T_2s+\frac{\sqrt{3}}{2}[P_1t_1+Q_1t_2-\tfrac{1}{2}(P_2t_1+Q_2t_2)]-\tfrac{3}{4}v_1S_2\\
&-\beta_2M^r_2+(\alpha_1+\beta_1)M^l_3-\tfrac{1}{2}(\alpha_1-\beta_1)M^r_3=-T_2s-\tfrac{1}{2}T_3s\\
&\quad-S_2v_2+\tfrac{1}{2}S_3v_1-\frac{\sqrt{3}}{2}(P_3t_1+Q_3t_2)\\
&-\alpha_2M^l_2+\tfrac{1}{2}\alpha_2M^r_2+\alpha_1M^l_3-(\tfrac{5}{4}\alpha_1+\tfrac{3}{4}\beta_1)M^r_3\\
&\quad=\tfrac{3}{4}T_3s+\frac{\sqrt{3}}{2}[P_2t_1+Q_2t_2-\tfrac{1}{2}(P_3t_1+Q_3t_2)]-\tfrac{3}{4}S_3v_1
\end{aligned}\right\}\ [\mathbf{32}]$$

37 *Crankshaft Subject to Twist by the Couples "T" Acting at the Ends.* Equations [**32**], written for the two intermediate supports, then reduce to:

$$\left.\begin{aligned}
(\alpha_1+\beta_1)M^l_2-\tfrac{1}{2}(\alpha_1-\beta_1)M^r_2-\tfrac{1}{2}(\alpha_2+\beta_2)M^l_3+\alpha_2M^r_3&=-\tfrac{3}{2}Ts\\
\alpha_1M^l_2-(\tfrac{5}{4}\alpha_1+\tfrac{3}{4}\beta_1)M^r_2+(\tfrac{3}{4}\beta_2-\tfrac{1}{4}\alpha_2)M^l_3+\tfrac{1}{2}\alpha_2M^r_3&=\tfrac{3}{4}Ts\\
-\beta_2M^r_2+(\alpha_1+\beta_1)M^l_3-\tfrac{1}{2}(\alpha_1-\beta_1)M^r_3&=-\tfrac{3}{2}Ts\\
-\alpha_2M^l_2+\tfrac{1}{2}\alpha_2M^r_2+\alpha_1M^l_3-(\tfrac{5}{4}\alpha_1+\tfrac{3}{4}\beta_1)M^r_3&=\tfrac{3}{4}Ts
\end{aligned}\right\}\ [\mathbf{33}]$$

38 As a numerical example the three-throw crankshaft of a 900-hp. Diesel engine will be considered, all dimensions of which are given in Par. 25. From these and Equations [**3**], [**6**], [**9**], [**10**], [**14**], [**18**], and [**19**] it is found that

$$\left.\begin{aligned}
&\alpha_1=0.851\,\frac{2e+f}{B_1};\quad \alpha_2=0.478\,\frac{2e+f}{B_1};\quad s=0.232\,\frac{2e+f}{B_1}\\
&\beta_1=0.864\,\frac{2e+f}{B_1};\quad \beta_2=0.381\,\frac{2e+f}{B_1};\quad v_1=0.408\,\frac{(2e+f)r}{B_1};\\
&v_2=0.640\,\frac{(2e+f)r}{B_1};\ t_1=0.559\,\frac{(2e+f)r}{B_1};\ t_2=1.013\,\frac{(2e+f)r}{B_1}
\end{aligned}\right\}\ [\mathbf{34}]$$

Applying these figures to the case discussed in Par. 37, represented by Equations [**33**],

$$\begin{aligned}
1.715M^l_2+0.0065M^r_2-0.429M^l_3+0.478M^r_3&=-0.348T\\
0.851M^l_2-1.712M^r_2+0.1663M^l_3+0.239M^r_3&=0.174T\\
-0.381M^r_2+1.715M^l_3+0.0065M^r_3&=-0.348T\\
-0.478M^l_2+0.239M^r_2+0.851M^l_3-1.712M^r_3&=0.174T
\end{aligned}$$

from which

$$M^l_2=-0.210T;\ M^r_2=-0.260T;\ M^l_3=-0.260T;\ M^r_3=-0.205T$$

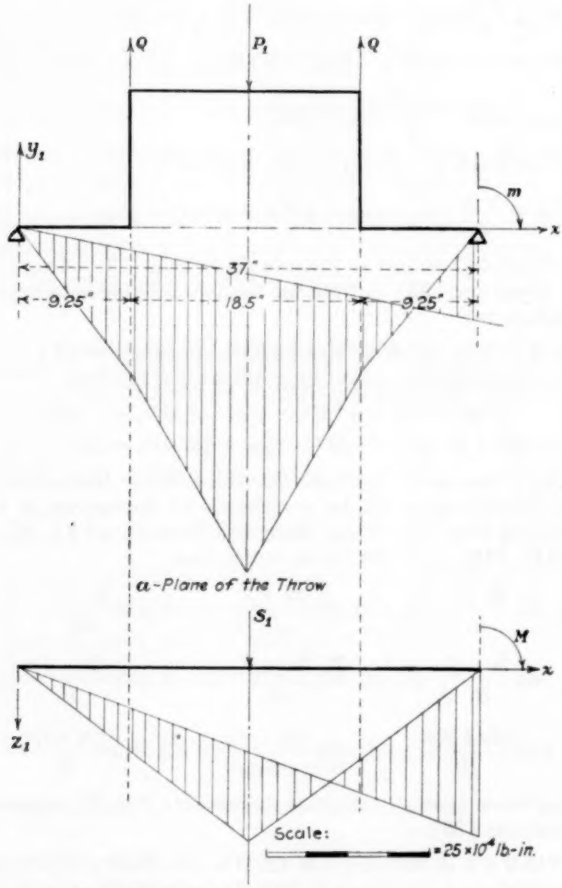
39 The angle of twist for any crank is now easily obtained. Equation [**12**] gives the twist if there is no constraint, namely,

$$\delta=Tw$$

The diminishing of this angle, due to the effects of the bending moments M, is calculated from Equation [**15a**]:

For the first throw,

$$\delta' = -0.210Ts$$



b-Plane Perpendicular to Plane of Throw.

Fig. 13

For the second throw,

$$\delta''_0 = -(0.260 + 0.260)Ts$$

For the third throw,

$$\delta'''_0 = -0.205Ts$$

The decrease of reduced length, due to the constraint, is:

For the first throw,

$$\frac{\delta'_0}{\delta} = \frac{0.210s}{w} = 0.0305, \text{ i.e., about 3 per cent.}$$

For the second throw,

$$\frac{\delta''_0}{\delta} = \frac{0.520s}{w} = 0.0775, \text{ i.e., about 7.8 per cent.}$$

For the third throw,

$$\frac{\delta'''_0}{\delta} = \frac{0.205s}{w} = 0.0301, \text{ i.e., about 3 per cent.}$$

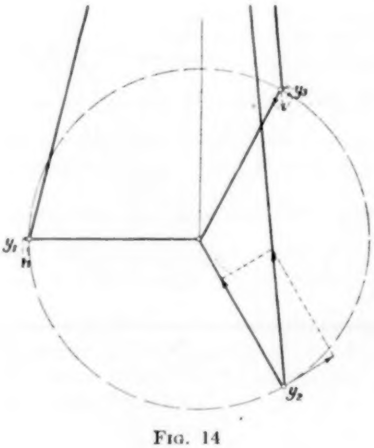


FIG. 14

40 In this manner the question of reduced length of the shaft considered is solved. It is seen that the effect of the constraint on the reduced length of the first and third throws is small. For the middle throw this effect is about half of that calculated in Par. 25 on the assumption that the throw is clamped at the middle of the journals. Similar results may be expected for any three-throw crankshaft with $\gamma_2 = \gamma_3 = 120$ deg.

41 The bending and twisting of the same crankshaft due to the forces P, Q, and S acting on the crankpins may now be considered. The forces P and S are obtainable from the gas pressure and inertia force diagrams. Examining first the case given in Fig. 12, where the maximum force acts on the first throw, the values of the forces considered are as follows:

Number of crank	Angle of crank, deg.	P	S	Q
1	20	− 65,000 lb.	29,400 lb.	7,700 lb.
2	140	− 9,400 lb.	− 5,830 lb.	7,700 lb.
3	260	− 4,300 lb.	11,100 lb.	7,700 lb.

Substituting in [**32**] the numerical values of [**34**] and the values of the forces given in the preceding table,

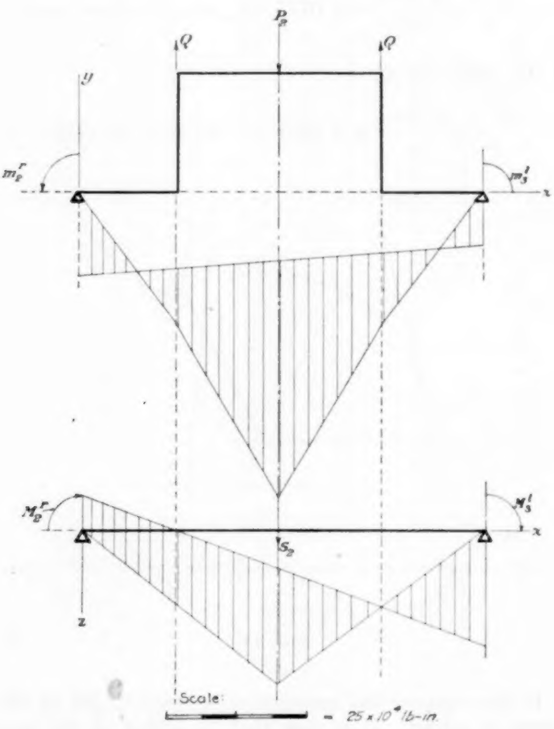


FIG. 15

$$
\begin{aligned}
1.715M^l_2 + 0.0065M^r_2 - 0.429M^l_3 + 0.478M^r_3 &= -399 \times 10^3 \text{ lb-in.}\\
0.851M^l_2 - 1.712M^r_2 + 0.1663M^l_3 + 0.239M^r_3 &= -206 \times 10^3 \text{ lb-in.}\\
-0.381M^r_2 + 1.715M^l_3 + 0.0065M^r_3 &= -\ \ 96 \times 10^3 \text{ lb-in.}\\
-0.478M^l_2 + 0.239M^r_2 + 0.851M^l_3 - 1.712M^r_3 &= 6.47 \times 10^3 \text{ lb-in.}
\end{aligned}
$$

from which

$$
M^l_2 = -258 \times 10^3 \text{ lb-in.} \qquad M^r_2 = -7.15 \times 10^3 \text{ lb-in.}
$$
$$
M^l_3 = -54.5 \times 10^3 \text{ lb-in.} \qquad M^r_3 = 40 \times 10^3 \text{ lb-in.}
$$

These values substituted in Equations [**28**] give

$$m^l_2 = -140 \times 10^3 \text{ lb-in.} \qquad m^r_2 = -287 \times 10^3 \text{ lb-in.}$$
$$m^l_3 = -77.5 \times 10^3 \text{ lb-in.} \qquad m^r_3 = -86 \times 10^3 \text{ lb-in.}$$

42 The diagrams of bending moment for the first crank, where the maximum bending moment is to be expected, are given in Fig. 13. The upper one gives the moments in the plane of the throw and the lower those in a plane perpendicular to the plane of the throw. The stress in any element of the throw may now be calculated in usual manner.

43 Secondly, consider the case shown in Fig. 14, where the maximum force is acting on the second crank. The corresponding values of forces, obtained from gas pressure and inertia force diagrams, are as follows:

Number of crank	Angle of crank, deg.	P	S	Q
1	90	3,030 lb.	13,250 lb.	7,700 lb.
2	210	− 70,000 lb.	30,600 lb.	7,700 lb.
3	330	− 8,100 lb.	− 5,970 lb.	7,700 lb.

Proceeding now as in Par. 41, Equations [**32**] give

$$M^l_2 = 101.5 \times 10^3 \text{ lb-in.} \qquad M^r_2 = -68.5 \times 10^3 \text{ lb-in.}$$
$$M^l_3 = -218 \times 10^3 \text{ lb-in.} \qquad M^r_3 = -23.8 \times 10^3 \text{ lb-in.}$$

which values, substituted in Equations [**28**], give

$$m^l_2 = 137 \times 10^3 \text{ lb-in.} \qquad m^r_2 = 157 \times 10^3 \text{ lb-in.}$$
$$m^l_3 = -98.3 \times 10^3 \text{ lb-in.} \qquad m^r_3 = -238 \times 10^3 \text{ lb-in.}$$

44 The greatest stresses are now to be expected in the second throw. The diagrams of bending moments for this throw are given in Fig. 15, from which the corresponding stresses can be calculated in the usual manner.

DISCUSSION

R. Eksergian. So far as the writer knows, this paper is the first complete solution of the multiple-throw crankshaft problem, though a partial solution has been made by Bach in his *Die Maschinen-Elemente.* In other works on design various very rough approximations have been made. The equations established in this paper, as well as the method of analysis, serve as a rational basis for the more approximate design formulas that would be used in practice. In establishing the equations of flexural rotation at the ends for any crank-throw panel the analytical work has been greatly simplified by the bending-moment-loading diagrams. These represent the distribution of $\int \frac{M}{EI} dx$ and are of the exact form

of those so frequently used for the simple beam fixed at the ends, the problem of three moments, etc.

It is necessary to point out the limitations of the equations in their practical application. The author assumes a crankpin diameter the same as the shaft. While the necessary modification for different diameters can be taken care of, its introduction may not be advisable, because of increased complexity in the equations. Level supports are assumed for the bearings, but in the continuous beam theory a small subsidence in one of the supports may cause large changes in bending moments and in their distribution. Hence it may be necessary in practice to introduce maximum assumed vertical displacements at the bearings. While this will complicate the equations, it is desirable, since with all the supports assumed level the bending moments will be too low, while the present equations, with one bearing assumed out of line in a multiple-bearing crankshaft, will give bending moments that are too high. It is probable, however, that the flexibility of the webs will permit a small subsidence of one bearing without causing much change in the values given by the present equations. Finally, the equations are applicable only to vertical rectangular webs.

The present method for the computation of multiple-throw crankshafts is usually based on a two-bearing crankshaft for a single throw simply supported at the bearings. The effect of torsion of the forward cranks is either added as a pure torsion moment, or as causing equal and opposite shearing reactions at the crankpin, the corresponding bearing reactions causing additional bending in the shaft. The reactions due to torsion for either half of the crank form a couple in a longitudinal plane at right angles to the crank, and this implies a moment constraint at the bearings. If this constraint exists, the bearing cannot consistently be considered as offering no bending restraint for the rod thrust loading. Thus even a consistent approximate formula must be based on the lines of this paper.

No. 1908

THE SOLID-INJECTION OIL ENGINE

By H. F. Shepherd,[1] Springfield, Ohio

Non-Member

The development of the solid-injection oil engine during the past three years has engaged the attention of many designers. Unfortunately, most of the information gained by experiments has been withheld from publication and the net result has been a duplication of laboratory and test-floor effort. The author has undertaken to present the most pertinent facts on the development of the engine from the former hot-surface unit and points out the influence of combustion-chamber design, spray angle and velocity, atomization, detonation, and other problems confronting the oil-engine designer.

THE development of the principles of solid injection has required a far longer period of time than did the development of the full-Diesel engine with air injection. It is safe to assume, however, that equal technical ability was brought to bear on both types, for Dr. Diesel first concerned himself with solid injection and abandoned it for what he considered good reasons. Rundlof first employed solid injection, then a compromise system using moderate air pressure, and later returned to solid injection. Jacob Gunther used air injection and finally developed a notably successful system of solid injection, while Hesselman, an international figure in Diesel-engine design, has lately produced an outstanding solid-injection model.

2 From these events it may be inferred that the design of the solid-injection engine has presented some difficult problems. Undeniably it has, but it is most pleasant to testify that progress has resulted in simplification of theories and practice. The modern solid-injection engine is as much a "rational heat motor" as the modern Diesel engine.

3 Rationalization of solid-injection-engine design has had to await the oil-engine man's education concerning the properties of petroleum, especially its behavior when subjected to heat, an influence less prominent when air injection is used. The experi-

[1] Consulting Engineer, Foos Gas Engine Co.

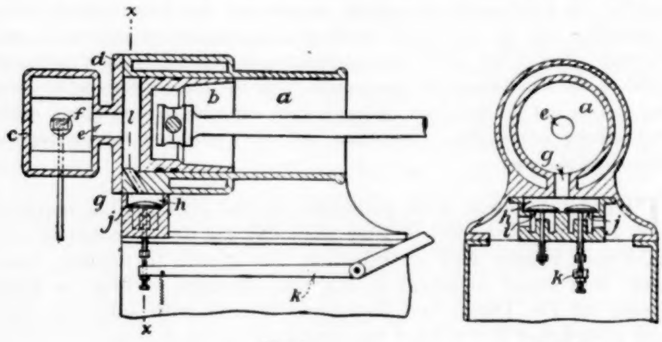
Contributed by Gas-Power Division and presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

menter had still further to extend his investigations into the hydraulic problems introduced by the injection system itself, and to extend his tests and observations in thermodynamics beyond that more familiar section dealing only with heat and work into a thorough study of conductivities, as will appear.

Historical

4 Herbert Akroyd Stuart proposed a solid-injection engine prior to Dr. Diesel's first experiments. His British pâtent No. 7146, dated May 8, 1890, describes

> . . . means for preventing the premature or pre-ignition of an explosive charge of combustible vapor of gas and air when a permanent igniter (such as a continuous spark or a highly heated igniting chamber) is in communication with the interior of the cylinder, by first of all compressing the



Figs. 1 and 2 Patent Drawings of Hornsby-Akroyd Engine

> necessary quantity of air for the charge and then introducing into this quantity of compressed air the necessary supply of combustible liquid, vapor or gas to produce the explosive mixture.

5 Unfortunately this engine was never known to commerce and Akroyd Stuart turned to a principle more in keeping with the low-pressure practice of that day. In British patent No. 15,994 A.D. 1890, he describes what we know as the Hornsby-Akroyd engine, which for some years, despite all its faults, was the best hot-surface engine available. In this latter patent specification he states:

> When liquid hydrocarbon is employed the injections may be so timed as to occur at the beginning or at any portion of the suction stroke or during the compression stroke. We prefer, however, to give sufficient time for complete vaporization in order to obtain the maximum economy of combustible liquid, and this can be accomplished by injecting the hydrocarbon during the suction stroke when the combustible liquid after coming into contact with the hot walls of the explosion chamber vaporizes very completely and occupies the combustion space.

6 The scheme of this engine is presented in Figs. 1 and 2 by means of the original patent drawings. Oil entered the hot chamber c through the spray f. The injection pump was driven by the downward stroke of the inlet-valve lever so that injection occurred during the first half of the admission stroke. The description proceeds:

No mixing sufficient to cause an explosion will take place owing to the contracted area of the communicating passage until, on the return or compression stroke . . . sufficient air is forced . . . into the explosion chamber to produce a mixture capable of explosion. The cubical contents of the engine cylinder and the explosion chamber are so proportioned that the mixture in the explosion chamber during compression is rendered explosive in time to secure the ignition at or near the end of the compression stroke.

7 In a measure the engine accomplished its object. It was a vast improvement over the external-vaporizer models of the day. At full load it gave a very creditable Otto-cycle card as shown in Fig. 3, but at lighter loads with smaller charges the relations were

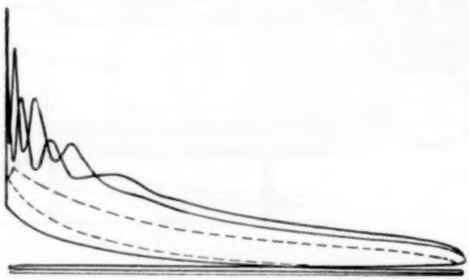


Fig. 3 Indicator Diagram of Hornsby-Akroyd Engine

destroyed; the explosive mixture was produced earlier and cards such as shown by the broken lines resulted. The isolation of the combustion chamber, serving to localize the carbon produced by excessive heat and prolonged "cooking" of the oil, was, in that day, a real virtue. Attempts made to rationalize the proportions of this engine, were, as far as the author knows, unsuccessful.

8 About 1913 Louis Doelling experimented along this line. He had an advanced knowledge of sprays, a better understanding of the requirements of the fuel pump, and possibly a full faith in the possibilities of the original Akroyd Stuart idea of injection at firing time. He reduced the clearance l, Fig. 1, by successive stages. Improvement resulted with each reduction. To eliminate the valve-pocket clearance h, the engine, Fig. 4, was designed with injection timed to the moment of firing, resulting in a new and greatly improved commercial engine, the De La Vergne D. H. Figs. 5 and 6 are full-load and quarter-load diagrams. In the course of further development the spray angle was gradually elevated with notable improvement, greatest economy resulting when the spray was

directed to meet the air entering the neck rather than when striking the hot surface directly in the old mode. The form of the hot surface was then altered as shown in Fig. 7.

9 Out of this, by the happy combination of two such slopes as shown in Fig. 4 and two such sprays, came the Price engine, Fig. 8. The convention date on the patent issued to the late Wm. Tudor Price is Dec. 10, 1917.

10 The dates succeeding the original Akroyd Stuart patents are not cited to show priority. Rundlof obtained rational working in the Bollinder engine before the appearance of the D. H. engine.

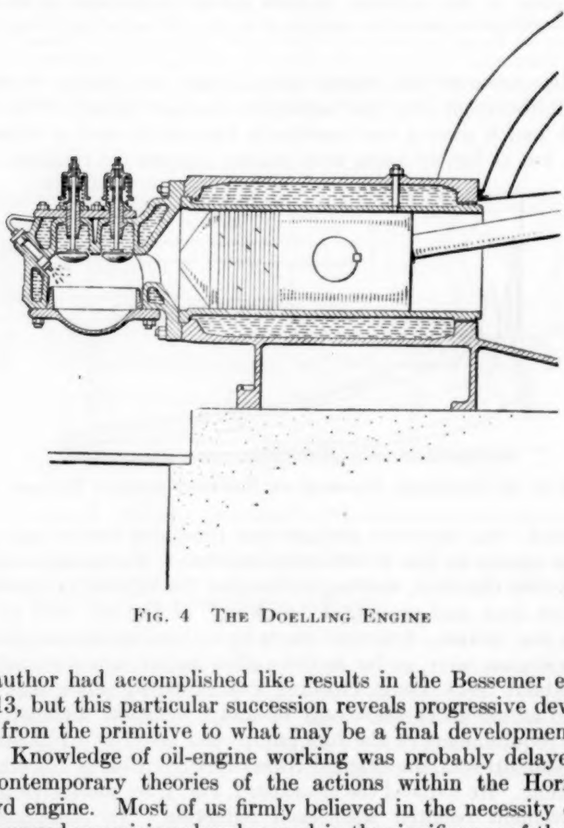


Fig. 4 The Doelling Engine

The author had accomplished like results in the Bessemer engine in 1913, but this particular succession reveals progressive development from the primitive to what may be a final development.

11 Knowledge of oil-engine working was probably delayed by the contemporary theories of the actions within the Hornsby-Akroyd engine. Most of us firmly believed in the necessity of an unscavenged vaporizing chamber and in the significance of the relative clearance inside and outside of the chamber, and had ideas on the action of the contracted passage dividing the combustion chamber into two parts.

12 The Doelling-Hornsby engine clearly showed improvement with its single-scavenged combustion cavity. The improved

Doelling-Hornsby engine demonstrated the secondary importance of the hot surface, and the Price engine, obviating the hot surface

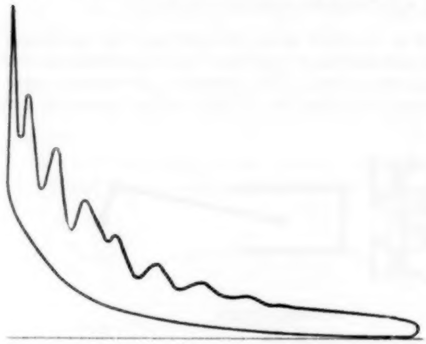


Fig. 5 Full-Load Diagram of Doelling Engine

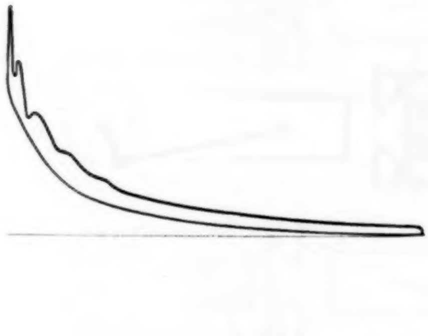


Fig. 6 Quarter-Load Diagram of Doelling Engine

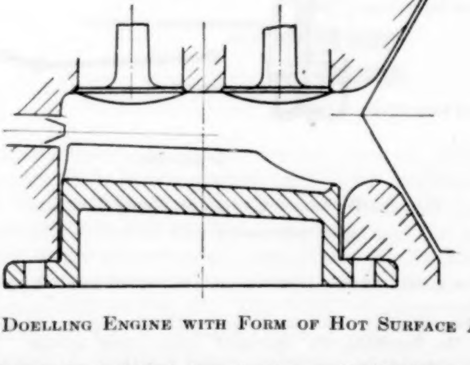


Fig. 7 Doelling Engine with Form of Hot Surface Altered

entirely, demonstrated that the air charge itself when compressed to the proper degree may perform the vaporizing function.

13 In January, 1894, Dr. Diesel had established the possibility of spontaneous ignition by injecting fuel into an atmosphere heated

by compression, and began experiments on the method of introducing fuel. A free translation from his book, Die Entstehung des Diesel-motors, shows his impressions seemed to be:

> Next, injection by means of the fuel pump direct, only an automatic valve in the nozzle, over the seat surface of which a conical distribution and atomization of the fuel shall take place. The operation is entirely unreliable. The degree of atomization observed under various pressures: at

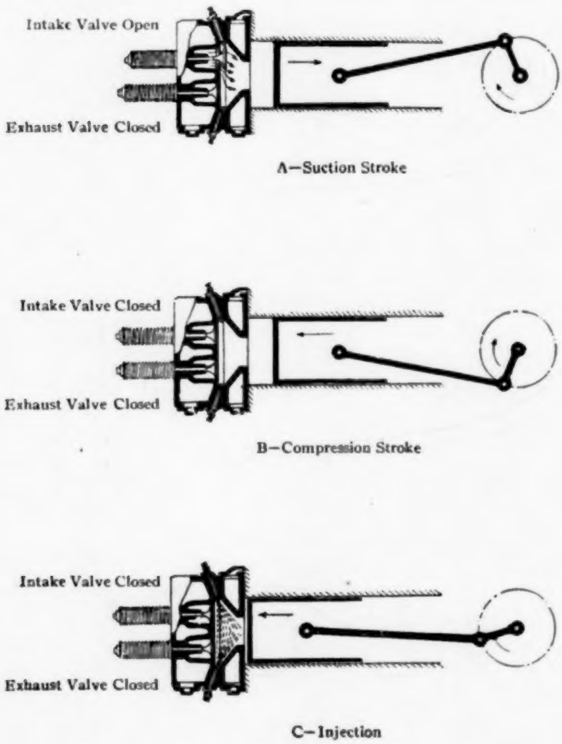


Fig. 8 Working Cycle of Price Engine

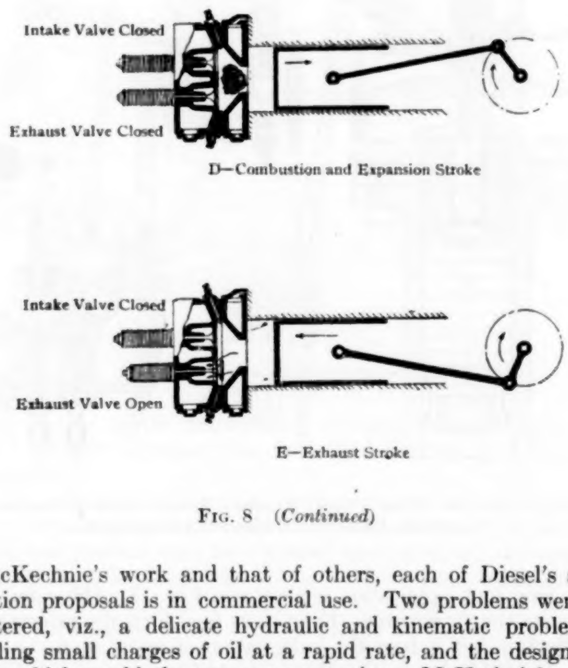
(See continuation on p. 477)

> high pressure good; at low pressure uninterrupted jets without the least atomization.
>
> Further experiments, injecting direct with the pump and simultaneous opening of the mechanically operated spray-valve needle. The pump plunger and spray-valve needle are linked together and operated by the same cam. Injection very precise.
>
> Next the pump was set aside and it was sought to accomplish injection by mechanical operation of the needle, the fuel pipe being kept under constant pressure. After these trials in the open air followed tests in operation with direct injection out of the fuel pressure pipe; ignition excellent. Ex-

haust gases burning still as they leave the cylinder, the spray valve is most unreliable, injection uncontrollable, the system must be abandoned.

14 Dr. Diesel's next step was to apply air injection with astounding and lasting success. Figs. 9 and 10 show the fuel pump, the various sprays, and the combustion chamber used in the experiments. The air chamber and check for the storage of injection air from the main cylinder were not part of his solid-injection apparatus.

15 Fifteen years later James McKechnie (Vickers & Maxim) undertook exactly the same series of experiments. As the result



D—Combustion and Expansion Stroke

E—Exhaust Stroke

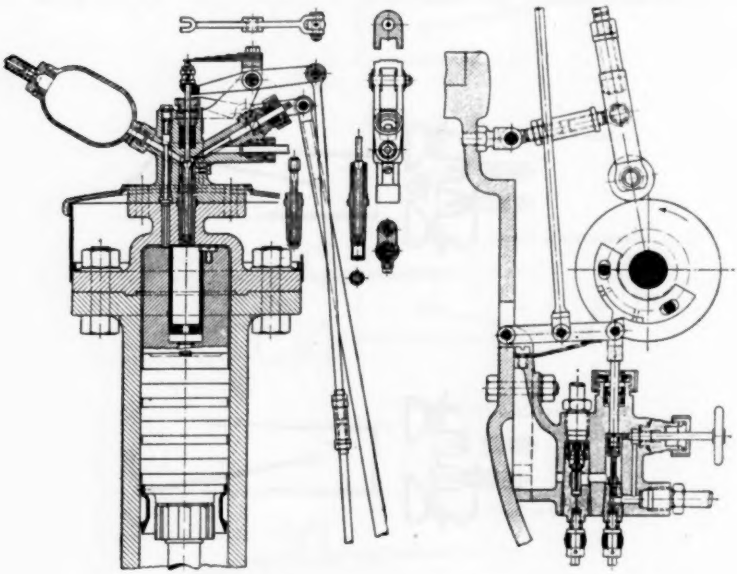
Fig. 8 (*Continued*)

of McKechnie's work and that of others, each of Diesel's solid-injection proposals is in commercial use. Two problems were encountered, viz., a delicate hydraulic and kinematic problem in handling small charges of oil at a rapid rate, and the design of a piston which would also serve as a vaporizer. McKechnie's earlier timed pump, accumulator (A), combustion chamber, and spray are shown schematically in Fig. 11. The later "common rail" system with a pump M and a pressure main P serving mechanically operated sprays g, shown schematically in Fig. 12, is the basis of the Vickers, Atlas Imperial, and Enterprise engines.

16 At first the production of injection air was considered a handicap and an airless system a desirable attainment. Among those seeking to avoid the use of a compressor Friedrich August

Haselwander came into prominence in Germany about 1901. He finally formulated the following claim:

In an internal-combustion engine the use of a division wall or recess or the like adapted to co-act with the piston in such a way that shortly before the end of the stroke the space in front of the piston is divided into two parts in which the air or gas is compressed at different rates as the piston advances, while arrangements are provided whereby the escape of air or gas at higher pressure into the chamber containing air or gas at lower pressure is utilized to inject and mix the combustible with the air.



FIGS. 9 AND 10 FUEL PUMP, VARIOUS SPRAYS, AND COMBUSTION CHAMBER USED IN DIESEL'S EXPERIMENTS

17 One such application is shown in Fig. 13. Oil is fed in at p^2 to meet the air streams. Even in its final form the principle was not effective but it held the imagination of many men.

18 In 1915 Jacob Gunther (Deutz-Otto) proposed the arrangement shown in Fig. 14, the oil entering at g. In 1916 he proposed that of Fig. 15, which was distinguished by both the point of injection and the conical neck and displacer. In his commercial engine shown in Fig. 16 the air jet is concentrated to oppose further advance of the spray. This was a highly efficient and satisfactory motor.

19 This incomplete historical sketch shows that oil-engine development, after the first basic proposals, was in reality not so

much a problem in invention as a labor of acquisition of facts and figures necessary to determine the proper proportions and arrangement of the devices originally suggested. It is now in order to sum up the individual problems.

Cracking

20 Cracking was known to refiners as early as 1861 and used as a process soon after. The engine trade began to give it intelligent and practical consideration, in 1903, when William John Crossley and Wilfred Le Plastrier Webb in a patent application covering the use of water injection in their particular mode, stated that "any liquid or vaporized hydrocarbons which come into contact with the overheated parts of the vaporizer are thereby

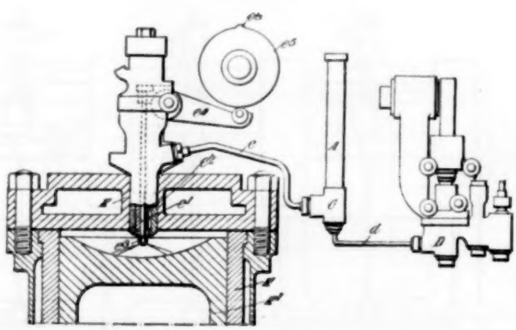


Fig. 11 McKechnie's Experimental Engine

decomposed or partly decomposed, forming carbon or tarry deposits." Water injection was then used to limit the vaporizer temperature.

21 In early engines the hot surfaces were invariably too hot, but by the use of water injection at different rates and economy tests it was learned that best results were attained at vaporizer temperatures below red heat. The author's final investigations were made on the cylinder head, Fig. 17, which proved to be a valuable laboratory device. Its principle is the control of the hot-surface temperature by the vaporization of a liquid (mercury) in contact with that surface. Excess heat is dissipated by the rise of mercury vapor to the condenser on top of the pot, the condensate returning to the well by gravity. By varying the pressure of air above the mercury, the boiling point of the liquid could be altered at will, and with it the temperature of the vaporizing surface. A thermoelement was staked into the bottom of the pot and led out to a pyrometer. Fuel economy and other characteristics were plotted against the hot-surface temperature. For a broad range of fuels the optimum temperature was very close to 850 deg.

fahr. Above this the decline of desirable characteristics was rapid. Little change took place between 680 and 850 deg. fahr.

22 From these results valuable factors were deduced. On the assumption that an overheated air charge due to excessive compression or incomplete scavenging would crack the fuel, the same as excessive vaporizer temperatures, limiting values for compression for various types of engines were estimated and successfully applied.[1]

Conductivities

23 In December, 1915, Mr. Louis Illmer contributed a paper on oil-engine-vaporizer proportions.[2] This paper marked the

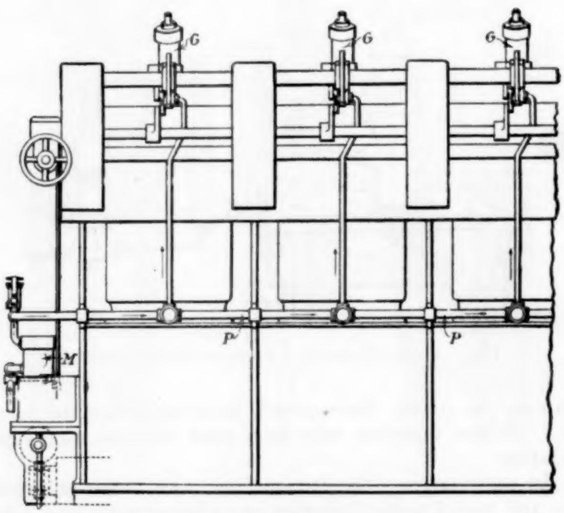


Fig. 12 Common-Rail System Serving as Basis of Vickers, Atlas, and Other Engines

passing of the Hornsby-Akroyd engine, but the section dealing with conductivities and the paths of heat flow remains most valuable in determining the hot walls of a combustion chamber within the temperature requirements. Thus, Par. 7 of this paper states:

The total temperature drop from the center to the edge of a thin disk (vaporizer) will equal the average drop per unit length multiplied by the disk radius, r, that is, for the disk

$$\theta = \theta_1 - \theta_0 \frac{1}{4} \frac{Hr^2}{KS}$$

[1] *Power*, vol. 57, p. 214. [2] Trans. A.S.M.E., vol. 37, p. 845.

where

θ_0 = temperature of disk at edge, deg. fahr.
θ_1 = temperature of disk at center, deg. fahr.
$\theta = \theta_1 - \theta_0$ = temperature drop
K = specific thermal conductivity
H = uniformly distributed gross heat input in B.t.u. per hr. per sq. in. of area
S = thickness of disk.

The external hot surface is now *passé*, but the piston head is a similar influence even in modern engines if the spray is directed against it, and must be treated in the same way.

24 The actual value of H is quite difficult to determine, but it is always the same for the full-load condition of any given type of engine. K is fixed for any given material, θ is fixed by practical imits. As regards the variables r and S, it is the author's custom

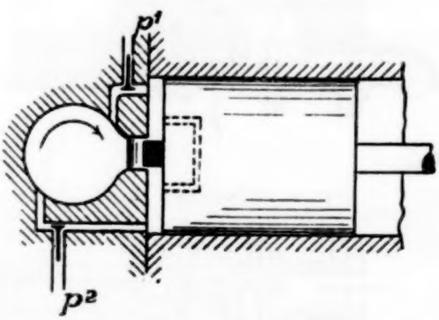


FIG. 13 HASELWANDER'S DIVISION WALL OR RECESS. OIL IS FED AT p^2 TO MEET THE AIR STREAMS

to fix S for one size by experiment and to vary it for the other sizes in the series according to r^2, thus solving the problem.

25 The Bristol Company made the author a contact thermocouple for use in this work. The engine is operated at full load until the exhaust temperature rises to a constant and fixed value signifying stable wall action. Preparation is made for a quick stop with the aid of the brake. A spray or valve is drawn in some fixed time, say, 30 sec., and the couple thrust against the piston surface. If the piston shows over 800 deg. fahr. it should be thickened. This applies to full-load operation. If the piston is too cold at light loads, some of the known means of compensation must be applied to increase the charge heat.

26 Realizing the importance of the piston-head temperature when the spray strikes it, the Ljusne Woxana Aktiebolaget actually fixed a thermocouple in the piston of a crosshead engine for piston-temperature readings in experimental operation.

27 A German patent (Paul H. Müller, No. 341,086, Sept. 11, 1920) describes an inspection opening created by the movement of

the inlet valve while the engine is in operation, which is especially adapted to the observation of the piston when it serves as a hot surface. See Fig. 18.

28 When the piston of a solid-injection engine serves as the vaporizer yet reaches such a size that cooling is required, a new application of the principles of conductivity is made. Since it is not feasible to spray against or along a jacketed piston wall any

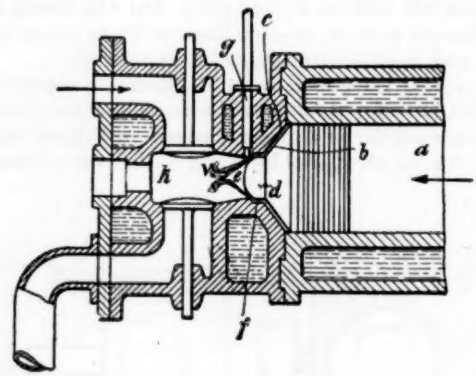


FIG. 14 GUNTHER'S ENGINE

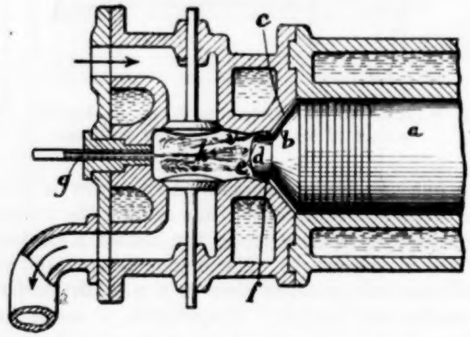


FIG. 15 GUNTHER'S ENGINE AS IMPROVED

more than against a jacketed cylinder head, a long path is provided for the heat flow from the piston-head surface to the piston jacket. Keller's scheme (Doxford) is shown in Fig. 19 and Regenbogen's (Krupp) in Fig. 20. Similar devices are found in true Diesel engines today.

DETONATION

29 The solid-injection engine working at its best is never a Diesel-cycle engine but is a mixed-cycle engine. The explosion

phase of the cycle serves to convey, disseminate, and intermix the clouds of vapor through the air.

30 Explosion is certainly a wave phenomenon, if not in the forced movement at least in the free movement before the restoration of equilibrium. The wave often shown on the indicator card is not a record of pressure, for if two or three springs are used in succession, waves of as many periods are recorded, and all agree

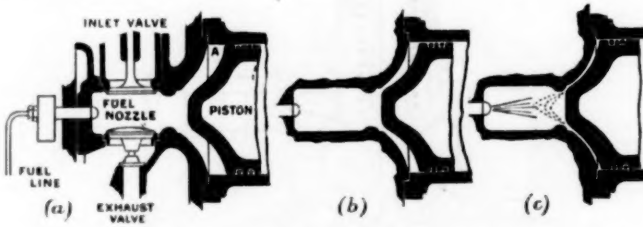


FIG. 16 GUNTHER'S ENGINE AS FINALLY BUILT

((a) Position of piston before air is partly trapped in ring-shaped space *A*. (*b*) Position of piston at start of air trapping. (*c*) Fuel injection at dead center in the air whorls.)

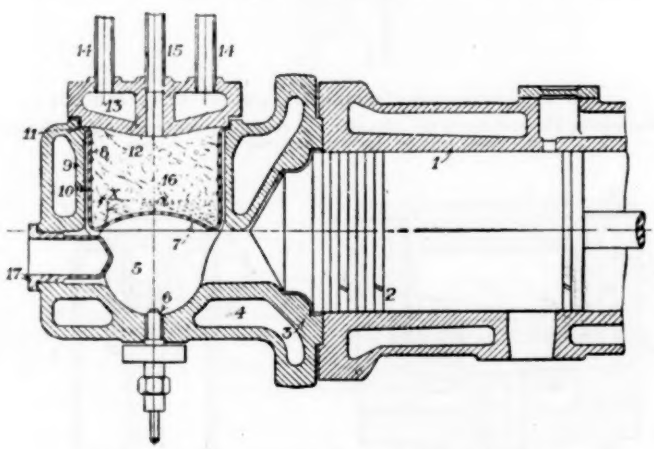


FIG. 17 EXPERIMENTAL HEAD USED BY AUTHOR

(In this head the hot-surface temperature is controlled by vaporization of a liquid in contact with the hot surface.)

with the natural period of the indicator motion for the spring in use.

31 The sound without the cylinder is certainly not transmitted from air waves within as there is no continuous medium. Various forms of cylinder heads give out various notes. Upon the sudden rise of pressure within, deflection takes place and then the walls vibrate until they come to rest. The piston has its own note, as

the inlet valve while the engine is in operation, which is especially adapted to the observation of the piston when it serves as a hot surface. See Fig. 18.

28 When the piston of a solid-injection engine serves as the vaporizer yet reaches such a size that cooling is required, a new application of the principles of conductivity is made. Since it is not feasible to spray against or along a jacketed piston wall any

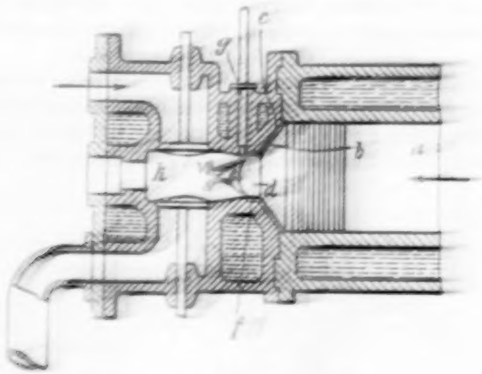


Fig. 14 Günther's Engine

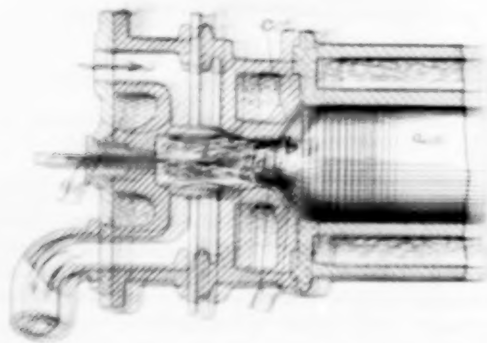


Fig. 15 Günther's Engine — [illegible]

more than against a jacketed cylinder-head, a long path is provided for the heat flow from the piston-head, a [illegible] in the piston [illegible]. [illegible] (Fig. 14 and Fig. 15) and [illegible] Fig. 20. Similar [illegible] Diesel engine [illegible].

OPERATION

29 The solid-injection engine working at its best is never a Diesel-cycle engine, but a mixed-cycle engine. The explosion

phase of the cycle serves to convey, disseminate, and intermix the clouds of vapor through the air.

30 Explosion is certainly a wave phenomenon, if not in the forced movement at least in the free movement before the restoration of equilibrium. The wave often shown on the indicator card is not a record of pressure, for if two or three springs are used in succession, waves of as many periods are recorded, and all agree

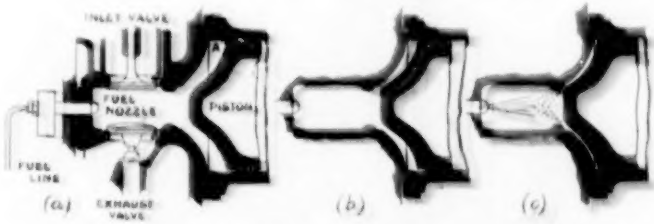


FIG. 16 GUNTHER'S ENGINE AS FINALLY BUILT

((a) Position of piston before air is partly trapped in ring-shaped space A. (b) Position of piston at start of air trapping. (c) Fuel injection at dead center in the air whorls.)

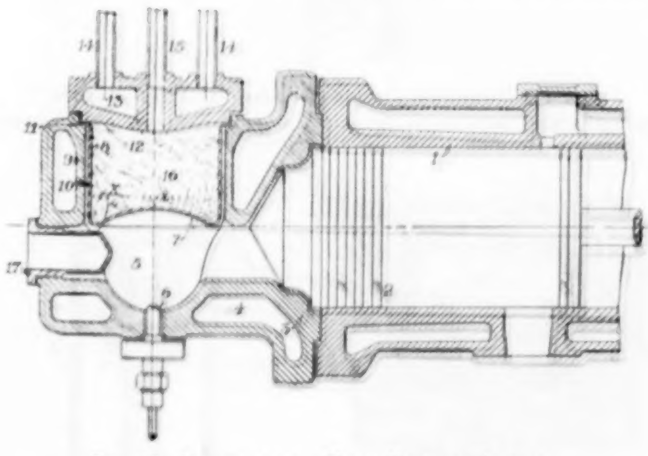


FIG. 17 EXPERIMENTAL HEAD USED BY AUTHOR

(In this head the hot-surface temperature is controlled by vaporization of a liquid in contact with the hot surface.)

with the natural period of the indicator motion for the spring in use.

31 The sound without the cylinder is certainly not transmitted from air waves within as there is no continuous medium. Various forms of cylinder heads give out various notes. Upon the sudden rise of pressure within, deflection takes place and then the walls vibrate until they come to rest. The piston has its own note, as

also has the crankshaft. Low-pressure engines with very heavy reciprocating parts, crossheads, etc., may be in a state approaching balance between inertia and compression, and the slack and the oil film may take up violently. High-compression engines

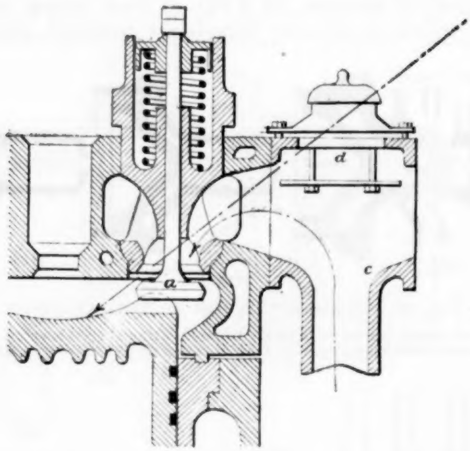


FIG. 18 INSPECTION OPENING DEVISED BY P. H. MÜLLER

(An opening d is provided in the inlet pipe c or the inlet-valve housing, through which the piston may be observed when the valve a is open.)

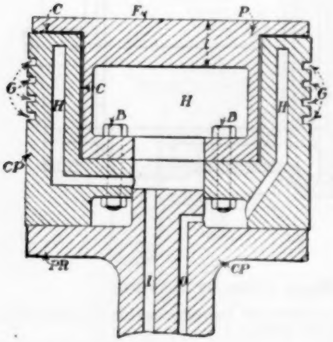


FIG. 19 KELLER'S SCHEME FOR COOLING PISTON SERVING AS VAPORIZER

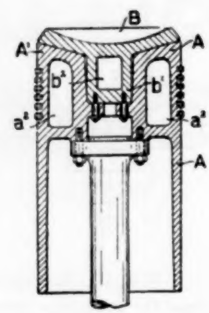


FIG. 20 REGENBOGEN'S SCHEME FOR COOLING PISTON SERVING AS VAPORIZER

with reasonably light reciprocating parts may be as quiet as gas engines and still give mixed-cycle indicator diagrams.

32 Not only does the explosion appear to be necessary, but the designer must consider the effect of the explosion upon the dissemination of vapor. If, in reacting to explosion, the vapor is

forced into contact with cold walls without travel through the air charge, the results will be bad. The vapor cloud may be arrested centrally with some admixture of air and driven radially by the explosion to good advantage. The vapor cloud may be a cone

Fig. 21 Right-Angle Diagram from an Engine Based on the Crossley Principle

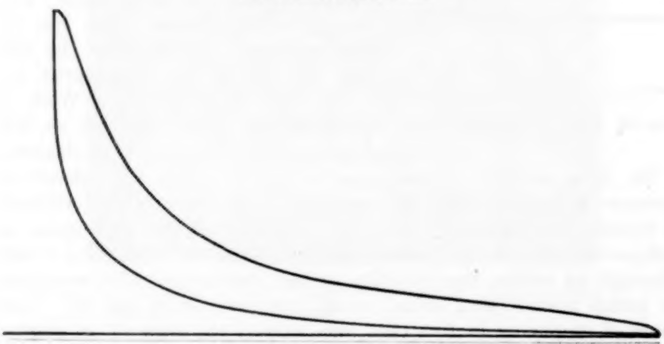


Fig. 22 Indicator Diagram from a Combustion Chamber Like that Shown in Fig. 11

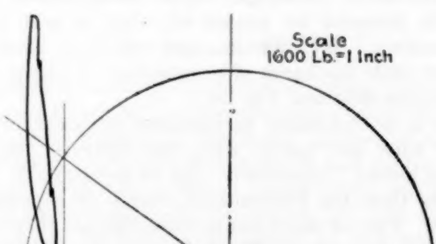


Fig. 23 Diagram for Modern High-Pressure Fuel Pump

with just enough air in its center to propel it outward through the air by a central explosion with excellent results.

33 Solid-injection engines today exhibit diagrams of most desirable characteristics. Fig. 21 is a right-angle diagram from an engine based on the Crossley principle. The pressure rise is by no means violent, yet it is smokeless. Fig. 22 is from a combustion chamber like that shown in Fig. 11 but with an automatic spray.

Hydraulics

34 The earliest fuel pump had every element of present-day systems. The multiple-orifice spray was used in the early Hornsby-Akroyd engine. The fuel bypass also was employed as a governing means. Yet these systems delivered fuel over a far longer interval than would be apparent from their mechanical timing.

35 The primitive diagram from a low-pressure pump handling a practically incompressible liquid is a rectangle. At present-day pressures of upward of 4000 lb. the elasticity of oil (about 0.00007 of the original volume per atmosphere) becomes a serious factor. The rise to pressure slopes, as also does the fall (Fig. 23). With an open spray an abrupt cam is necessary. With a spring-loaded spray the rise to lift pressure may be as gentle as desired.

Fig. 24 Spray-Pipe Diagram

36 If a suitably proportioned bypass is used, the release of pressure is rapid enough. If a variable-stroke pump is used without a bypass, the release of spray pressure through the open spray is subject to laws similar to those governing the emptying of a vessel through an orifice, the quantity of fuel discharged after cessation of pump travel being equal to the compression of the oil. The time of emptying in this case may exceed the time of an engine cycle. With a spring-loaded spray, cessation of discharge follows the law of discharge from a standpipe between levels, i.e., a very high and attenuated standpipe. The spring-loaded automatic spray may be designed by proper selection of seat areas for a known "blowdown," say, 500 lb., and will thus close promptly and with very little discharge after cessation of pump movement. See the spray-pipe diagram, Fig. 24.

37 Today it is customary to calculate the clearance in pump and delivery tube and spray, when the delivery due to elastic discharge at a known "blowdown" can be determined. This must always be less than the friction-load charge or a peculiar action will be set up. Two or more pump deliveries are then required to raise the pressure enough to lift the spray, after which the spray discharges the compressed volume and the operation is repeated. Such action, not uncommon, is avoided by proper consideration of the volume of oil acted upon relative to the required charge.

38 The hydraulic indicator is invaluable in investigating fuel-pump and spray phenomena. The indicator-piston displacement must be very small to avoid errors in deduction.

Atomization

39 In the solid-injection engine there is no means of producing an oil fog of such perfection as is produced by air injection. The minute drops of oil are not carried in on an energized vehicle, but must by virtue of their own mass and velocity carry from the nozzle to the desired location. In spray action, delivery accurate as to time, duration, and velocity is more important than atomization.

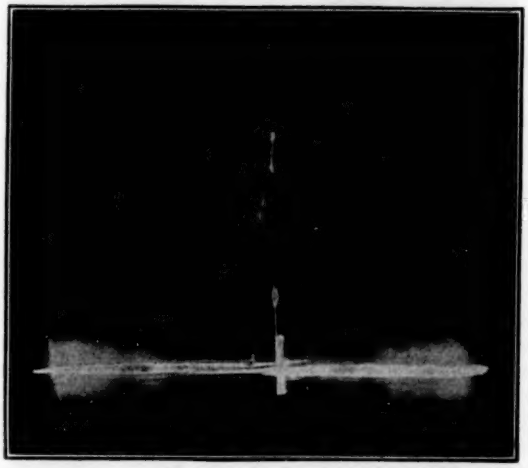


Fig. 25 Photograph of Needle Movement (Multiplied) in a Modern High-Pressure Automatic Spray Valve

Fig. 25 is a photographed record of the needle movement (much multiplied) in a modern high-pressure automatic spray valve. For promptness of action it leaves nothing to be desired.

Penetration

40 The preliminary development of sprays is usually carried out at the test bench with a power-driven pump. Here the penetration or carrying range of the spray is often a matter of feet rather than inches. This is particularly true of the multiple drilled-orifice sprays. The sprays with a single orifice and a spiral whorl to impart a rotary motion to the issuing jet, may be arranged to deliver a variety of profiles depending on the degree of rotary motion imparted by the tangent passages in the whorl and the diameter and length of orifice. This type of spray applies well to combustion chambers in which there is no hot surface, such as a

hot plate or piston head, to halt the travel of the fuel. The centrifugal action may be made to absorb most of the energy of introduction. Its cone of distribution is too narrow for Diesel-type combustion chambers in which multiple-orifice sprays are generally used.

41 There is much argument as to whether the jets from high-pressure multiple-orifice sprays actually penetrate the dense at-

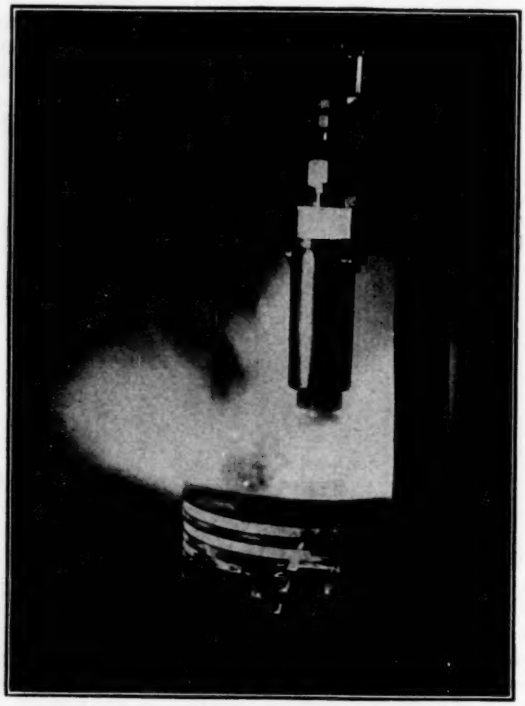


FIG. 26 MODERN SPRAY DISCHARGING AGAINST A PISTON HEAD HELD ARTIFICIALLY AT WORKING HEAT

mosphere of the combustion space and reach the opposite wall. If the sprays are operated under water, a medium far more dense and viscous, it will be found usually that they are capable of penetration in that element exceeding the travel in the combustion space. The author has caused the spray from a 0.012-in. orifice at 4000 lb. pressure to impinge upon the piston of a Hopkinson indicator at a distance of 3 in. and found 89 per cent of its energy recorded on the instrument. It is therefore almost true to Torricelli's law at short range. Apparently it is still necessary to look to vaporization for success in solid-injection engines rather than

to atomization, which for drilled sprays seems to be largely a surface effect.

42 Analysis of a number of recorded cases and of the author's own careful experiments on two high-pressure types seems to show that when the common arrangement of five spray holes is used in the tip, their diameter may be best determined by assuming a time of injection of, say, 0.01 sec. for moderate-speed engines and calculating the holes by the law $V = \sqrt{2gh}$, to which a suitable orifice coefficient is applied. Any attempt to atomize better by using smaller holes at any given pressure must only result in unduly extending the time of injection.

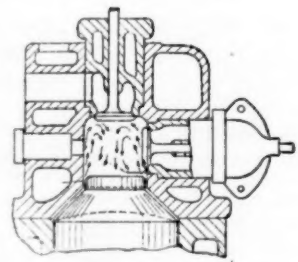


Fig. 27 Crossley-Type Combustion Chamber

Vaporization

43 It is the author's firm impression, checked by searching tests of no less than a dozen distinct and individual combustion-chamber arrangements, that creating proper conditions for vaporization is one of the main functions of a solid-injection system. Fig. 26, from a photograph taken during the conduct of an extensive search into the properties of sprays at the shops of the Foos Gas Engine Company, shows a modern spray discharging against a piston head held artificially at working heat. The hot air charge has much the same effect on the spray. Diffusion of these vapor clouds could not take place in the time of a combustion interval, so dependence is placed upon explosion to do the mixing.

44 Any one who has ever distilled petroleum in an Engler flask knows that even the compression and hot-surface temperatures will not fully vaporize most fuels. At any rate, to attempt to do so would result in cracking. The shattering effect of the explosion phase of the cycle undoubtedly atomizes the residues.

45 The terrific velocity of the jets acting in Fig. 11 results in vaporization far superior to that of the primitive hot-surface engine. The spray strikes at an angle of incidence which results in a scouring action, in imparting velocity to the vapor, and in a high rate of heat transfer to the fuel.

Ignition

46 The combustion-chamber air or wall conditions must be maintained such that inflammation does not take place instantly upon injection. This momentary delay is aided by the incomplete state of compression at the beginning of injection and by the

latent heat of vaporization of the fuel. If inflammation of the minute drops were to occur before vaporization, the smoky flame of a body of oil burning upon the surface would result.

Turbulence

47 Much inventive ability has been expended to produce artificial turbulence in the solid-injection engine. Engines of the Vickers type develop essentially the same economies as those fitted with displacers on the piston, etc. Differentiating the displacement of air through the neck between the cylinder and combustion space proper generally shows very low velocities. Even

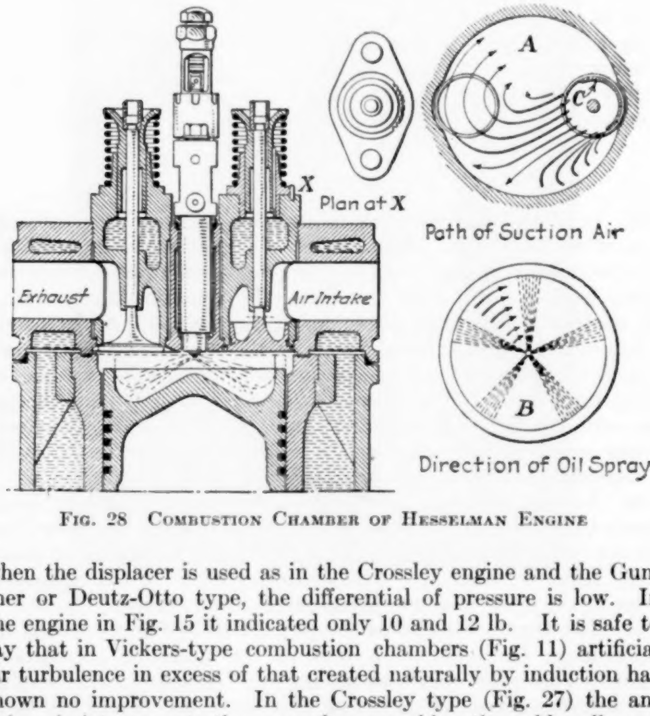


Fig. 28 Combustion Chamber of Hesselman Engine

when the displacer is used as in the Crossley engine and the Gunther or Deutz-Otto type, the differential of pressure is low. In the engine in Fig. 15 it indicated only 10 and 12 lb. It is safe to say that in Vickers-type combustion chambers (Fig. 11) artificial air turbulence in excess of that created naturally by induction has shown no improvement. In the Crossley type (Fig. 27) the annular air jet prevents the spray from reaching the cold walls and confines and returns the vapors to the core of the charge prior to explosion. In the Deutz-Otto type (Fig. 16) it checks overtravel of spray to the hot displacer. In both of these latter it serves to bring up fresh supplies of hot air for the vaporization of fuel.

48 The Hesselman case (Fig. 28) appears to be distinct, though opportunity has not yet been afforded to check it experimentally. Professor Hesselman reports marked improvement by the use of

artificial means of creating turbulence, but the ultimate economy is not essentially better than in other engines of the same compression and spray pressure. It will be noted that the spray is directed to the circumference of the combustion chamber. Undoubtedly the explosion phase of the cycle is then not so efficient in scattering the vapors as it would be if the charge were more central and adapted to radial and outward dissemination through the unsaturated air by explosion. In this case rotary turbulence is undoubtedly of value.

49 When no hot surface is present, progressive movement of the heated air in the path of the spray is very important. The specific heat of air is low and the mass of the air charge is small. If the spray is merely suspended, conduction is slow. For proper vaporization, then, the air should be kept moving. There is little such movement in Fig. 11, where it is not required. In Fig. 8 the volume of air outside of the combustion chamber at the beginning of injection approximately equals that inside. By displacement, the outer volume is moved up, bringing new supplies of heat.

Conclusion

50 The development of the solid-injection engine has suffered much by reason of the transmission of honestly proposed but fallacious theories of working. It has suffered, too, by a persistent desire to produce in an essentially individual type the characteristics of the air-injection Diesel. However, no single step in the development is without its contribution to the real theory of its working, and at this hour it would seem that the ultimate types have emerged from a seeming chaos of ideas to stand the test of time and progress.

51 Particular thanks are due Mr. S. F. Whitesel for his conscientious and painstaking assistance in the development of the evidences of spray action recorded in this paper, as well as for his photography.

DISCUSSION

P. H. Schweitzer.[1] The author has well expressed the present views and principles of solid-injection engine design. Progress has been very distinct in the last decade. The success of a solid-injection engine depends upon so many variables that one cannot but wonder that the pioneer experimenters attributed the failure of certain types to the failure of the principles involved rather than to some minor circumstances that were the real cause of failure.

Usually the historical development of a machine is selective.

[1] Asst. Prof. of Engineering Research, State College, Pa.

From a large number of types one or two survive, the others being discarded as inefficient. On the other hand, the development of the solid-injection oil engine has been quite different. There are more types now than ever, which shows that the old ideas and restrictions were wrong. But it should be clear that the dozen different types and principles cannot be equally good and efficient. One or two among them, based on superior principles, ultimately should be the most successful.

The theories and opinions concerning combustion are far from being uniform. The Price engine is built with two spray nozzles opposite each other on the sides of the cylinder, with the idea that the two sprays meet in the middle of the chamber, producing a good mixture and combustion. The engine is a success. Lately, the Falk engine with a 19-in. bore has been built on this principle, and the company emphasizes this action of breaking up the spray. The sprays have to travel 9½ in. before meeting. Hesselman calculated that an oil spray under similar conditions in compressed air does not penetrate more than ⅝ in., irrespective of the initial oil pressure. He built a successful engine on this principle, making the air to rotate in order to deliver the necessary oxygen particles to every oil particle. The author, as a result of his experiments in air and water, is of the opinion that the spray is able to penetrate a few inches at any rate. The writer cannot agree with him on the assumption that water offers more resistance than compressed air to the spray. Positive knowledge regarding the behavior of oil spray in compressed air should be obtained.

It is a fair assumption that *atomization* alone, if it is carried far enough, is sufficient to cause combustion. The true Diesel engine depends almost entirely on atomization. It has been observed that the preheating of the injection air caused detonation. In a solid-injection engine if the atomization is carried to this extent, using many small holes in the nozzle, the spray will not have the required energy to penetrate deep enough to meet the necessary number of oxygen particles. With more than five holes, we have unfavorable results. For the same reason the use of spiral grooves should be discouraged.

In the author's opinion we should rely more upon *vaporization*. At the German Diesel convention in 1923 Alt expressed doubt concerning the part that vaporization plays in combustion. His argument is that some oils have an ignition temperature many hundred degrees higher than their boiling point. The ignition temperature of some other oils is equal to, or somewhat below, the boiling point. If vaporization is essential for combustion, we should expect better combustion with oils whose ignition temperature is much above the boiling point, since the oil globules then have more opportunity to mix. No such result has been obtained. While Alt was dealing with the Diesel-type engine, the same reasoning can be applied, to a certain extent, to the solid-injection engine.

It is known that the ignition point is determined mainly by the

temperature and that compression has little effect upon it. Tizard advocated the theory that the ignition temperature is that at which the slow combustion of the fuel vapor with air evolves heat at a slightly greater rate than it is being dissipated by conduction. Hence ignition temperature depends on the rate of loss of heat and is different in turbulent air than in stagnant air. Experiments with and without fans show differences in ignition temperature up to 120 deg. fahr. This checks with the observation that the ignition temperature in the engine is generally higher than that measured outside.

Indicator diagrams of solid-injection engines show combustion of an explosive nature. The author considers this advantageous and essential for the dissemination of vapor. The writer believes that a flatter combustion line would be desirable provided it would not lead to after-burning. A better control of the fuel injection as to spray action and accurate delivery in quantity and timing might lead to this end.

There may be some doubt regarding the author's contention that an ordinary explosion is a wave phenomenon. The waves of indicator cards, as the author remarks, show different frequencies for different stiffnesses of springs. Cards taken from airplane engines, with refined methods, show rapid explosions but no waves. The indicators now in general use are not adapted to explosive engines, even for speeds not exceeding 500 r.p.m.

It is difficult to agree with the author that the external hot-surface is now *passé*. A successful low-pressure engine recently developed by the Venn-Severin Machine Company has an uncooled ring between the cylinder and head. The axial spray, which has an adjustable cone angle, strikes this external hot ring at starting, but after the engine becomes warm, the spray cone closes and no longer impinges upon the hot surface.

P. P. Bourne. The section of the paper entitled Detonation is especially interesting as dealing with phenomena existing in engines described by the author as "mixed cycle." It is generally understood that in this type of engine the pressure rises considerably during combustion, this rise being approximately equal to one-half the combustion pressure. Some of the better-known engines of this class are the Vickers, Crossley, Deutz, and engines built according to Price, such as the De La Vergne and Ingersoll-Rand. With such combustion the engine does not work upon the Diesel cycle, but the combustion more nearly corresponds to that of the explosive cycle. The high combustion pressure is taken care of in the design by providing parts with the necessary additional strength. However, this type of engine works under the disadvantage referred to by the author as a tendency to produce a detonation which imposes shock on the structure. It is undeniable that it is desirable to avoid this by obtaining a true Diesel cycle by solid injection.

The writer agrees in general with the author's statements con-

cerning detonation, but takes exception to the statement that the solid-injection engine, working at its best, is never a Diesel cycle engine, but a mixed-cycle one. One type of engine is built which works on a true Diesel cycle with solid injection. This engine uses what may be described as two-stage combustion, and the type in most common use in the United States is that developed and manufactured by the Worthington Pump & Machinery Corporation.

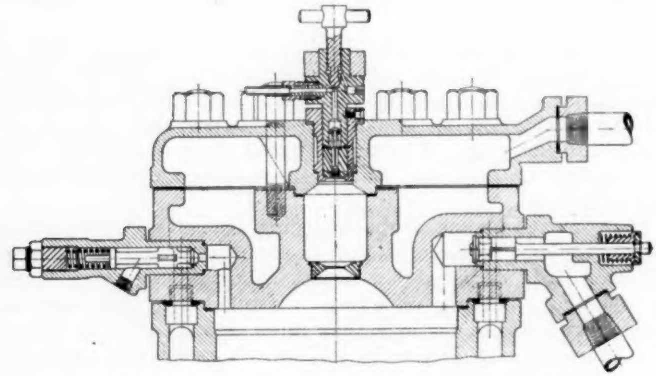


FIG. 29 CYLINDER-HEAD CONSTRUCTION OF TWO-STAGE COMBUSTION TYPE OF ENGINE

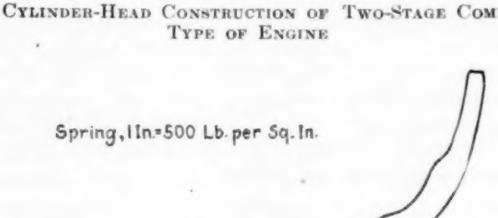


FIG. 30 TYPICAL INDICATOR CARD FROM TWO-STAGE COMBUSTION TYPE OF ENGINE

The construction, as shown in Fig. 29, indicates the combustion method. Direct pump injection is used at an oil pressure of not over 1000 lb. per sq. in. The fuel is injected into a precombustion chamber, where some of the oil is partially burned. The pressure rise in this chamber produces a pressure differential above the combustion pressure in the cylinder or combustion chamber proper, which causes the flow of a charge of burning fuel and gas into the cylinder at a rate sufficiently slow to produce the same combustion conditions in the cylinder as would exist in an air-injection Diesel engine. Fig. 30 shows a characteristic indicator card taken from an engine of this general type.

The construction of the cylinder head is such that all parts of the combustion and precombustion chambers, as well as the sprayer, are thoroughly cooled. The piston may be cooled to any extent desired without interfering with perfect performance. The larger sizes are fitted with oil-cooled pistons which work well in spite of the difficulty described by the author in a statement that it is not feasible to spray against or along a jacketed piston wall any more than against a jacketed cylinder head, probably because the fuel in this case is not in liquid form.

In this type of engine the distribution of the fuel through the air in the combustion chamber is attained by utilizing the high velocity of the gases through the throat between the two combustion chambers. Hence it is unnecessary to impart extreme energy to the fuel by injection at high pressure, and a simple form of injection apparatus becomes possible. The fuel consumption averages 0.33 lb. per indicated hp. per hr. The low injection pressure allows larger holes to be used in the spray nozzle, so that this is less delicate than would be otherwise possible. The openings are less likely to clog in operation, especially as the tip cannot become overheated.

Louis Illmer. This paper presents the subject of oil-engine research in a way that throws new light upon certain phases of it. It would be interesting to learn the views of the author as to the relative advantage of the types whose constructive features are discussed in the first portion of the paper, which the author may have tried out. This might be supplemented by the author's opinion as to the basic principles that should underlie good oil-engine design and the means preferred to attain this end.

In the latter portion of the paper the subject of fuel pumps might have been stressed to a greater extent. This accessory constitutes the heart of the direct-injection oil engine and represents the primary means of controlling the combustion process. Certain figures are given by the author relating to the elasticity of fuel oil, but no mention is made of the fact that the air entrained in the liquid may produce disturbances in pump action that far exceed those resulting from elasticity of the liquid proper. This air is further augmented by such additional air as may be trapped in the pump parts and discharge pipe. In smaller high-speed fuel pumps these combined air effects seriously interfere with the proper functioning of the injection process, and in extreme cases may cause continued injection of fuel throughout the expansion stroke.

The maintenance of high pressure behind the nozzle during the entire period of fuel injection is a most effective means for obtaining an extremely fine atomized condition of the fuel. Such surfacing is essential for rapid and perfect mixing of the fuel with all of the available air. Unless suitable means are provided to meet this need, the expected results will not be competitive with those obtained by air-injection methods.

Effective turbulence is also necessary to drive the finely divided particles away from the nozzle and cause them to be uniformly mixed throughout the available air. Perfect atomization, combined with ample penetration of the fuel, are believed to be essential for smokeless and economical combustion in airless-injection oil engines. Fine atomization, unaccompanied by adequate turbulence, will result in pocketed or localized combustion, no matter how thoroughly the fuel may be broken up or surfaced.

The writer has found the jet nozzle to give best results as regards fuel economy and maximum output per unit of piston displacement. The jet should impinge on the hot surface of the piston. When properly arranged it is possible to make the highly atomized fuel charge penetrate and thoroughly mix with all parts of the available air. Such injected fuel ignites almost instantaneously in a manner similar to that of a full-Diesel engine.

The size of nozzle hole is important in the satisfactory operation of direct-injection oil engines. A minimum diameter of $\frac{1}{64}$ in. is necessary to prevent clogging the nozzle. It is the writer's experience that a fuel-pump pressure of from 3000 to 5000 lb. per sq. in. is required to obtain satisfactory atomization and penetration of the more common grades of fuel oils.

Furthermore, it is important to maintain a correct ratio of air in which to burn a given injection of atomized fuel. These and similar points were covered in the writer's paper entitled Porting and Charging of Two-Stroke Oil Engines.[1]

Another important point is the prevention of dribble at the nozzle after the injection period terminates. For best results the pressure behind the nozzle should drop almost instantly to atmospheric pressure by venting the nozzle or discharge pipe back into a relieved pump chamber. Unless the pipe line and the entire system are sharply relieved of pressure, the gradual expansion of air in the oil discharge passage is likely to cause some of the fuel oil to dribble into the expanding cylinder gases.

S. A. Hadley. The author nowhere touches on the question of suitable fuels for the solid-injection engine. It would be interesting to know what he considers are the characteristics of a suitable fuel, its viscosity limits, and the results of preheating the fuel to various temperatures.

A rather extensive observation of various types of solid-injection engines seems to show that this type of engine has a much narrower range for suitable fuels than has the air-injection engine. Further discussion of this point would be very interesting.

[1] Trans., A.S.M.E., vol. 43, p. 649.

No. 1909

THE ECONOMICAL THICKNESS OF INSULATION IN REFRIGERATOR CARS[1]

By Arthur J. Wood,[2] State College, Pa.
Member of the Society

and

Philip X. Rice,[3] Danville, Ill.

The authors call attention to the increasing importance of the economic design of refrigerator cars and discuss the factors which must be taken into account in finding the proper thickness of insulation. The outstanding feature of the paper is the use of alignment charts to determine readily answers to problems involving many variables. By aid of these, or similar charts, one may determine not only the economic thickness of insulation but also the saving which would result from the adoption of the correct thickness compared with any other thickness; furthermore, the economic advantage of one kind of insulation over another may be found.

The paper gives a brief discussion on the economy of using regranulated cork or air spaces in some constructions. It outlines, by aid of average shade-temperature maps, the means of approximating external surface temperatures and notes the results of tests on roofs, sides, and floors as affected by direct sunlight and color of paint.

Reference is also made to the analysis by the authors of the cost of transporting extra weight of insulation, or extra weight of cars as a whole, and the final average figure is found to be about 0.10 *cent per ton-mile for transcontinental routes.*

All of the above and other factors form a part of the data for a complete solution for economic thickness, whether it is solved by a formula or by charts.

[1] This paper will constitute one of the chapters in a Bulletin to be issued by the Engineering Experiment Station, The Pennsylvania State College.
[2] Head of Department of Mechanical Engineering, Pennsylvania State College.
[3] Electrical Engineer, Miller Train Control Corporation.

Presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

THIS paper aims to explain a reasonably simple method for determining quickly and accurately the most economical thickness of insulation, and further, to discuss some of the many factors which enter into the question of the economical design of insulated walls. Although this discussion is concerned with refrigerator cars, the possibilities of the method, modified to meet the individual requirements, appear to be unlimited.

2 In a paper by Mr. P. Nicholls presented at the joint meeting of the A.S.M.E. and A.S.R.E. in December, 1922, there was developed a formula for solving problems similar to the one in question. He pointed out that trial calculations are laborious and unsatisfactory and that a better plan would be to develop a general equation which included all the factors in determining the economical thickness of insulation. The authors pointed out some of the practical limitations in applying an equation containing many variables and proposed a simple method requiring the use of charts. Rectangular coördinates may be used to advantage where there are but few factors, but are not so well adapted for complex relations; in which case alignment charts, as here shown, reduce the difficulties to a minimum.

3 If the trend in the present development is an indication, the refrigerator car of yesterday will not be the refrigerator car of tomorrow. The steam locomotive has passed through a similar transition; when wages, fuel costs and operating expenses were low, there was no great incentive to obtain the greatest economy and highest efficiency. In somewhat the same way when the prices of perishable food products were low and the supply abundant, the question of accurately accounting for the heat losses in refrigerator cars or of the spoilage of some of the products was not a matter of very serious concern. Not so today. The higher cost and greater demand have resulted in the analysis of costs which heretofore were regarded as unimportant. Furthermore the better understanding of the laws of heat transmission and the determination of trustworthy constants have brought about more rational methods of design.[1] The study of the authors indicates that nearly all of the refrigerator cars in the United States are insufficiently insulated. Until recently, papers dealing with an analysis of the entire problem have not been published.

4 The saving of perishable food and the reduction of the amount of ice required for that purpose, must be balanced against the various factors to accomplish this saving. Among many items which the authors consider are the percentage of time cars are in service, cost of transporting extra weight of insulation, depreciation of the cars, required amount of insulation for a

[1] An illustration in point is the design of the new refrigerator cars for the C.M. & St. P. Ry. described in *Railway Review*, June 16, 1923. See also Some Notes on Railway Refrigerator Cars, by W. H. Winterrowd, Trans., A.S.M.E., Vol. 44, p. 125.

particular structural design, effect of radiation, and the relative amount of absorption of heat due to different colors of paint. These and other factors must all be coördinated and the solution made simple and direct.

The Factors

5 The following factors enter into the determination of the economic thickness of insulation of a refrigerator car:

Investment. Fixed charges due to ownership of insulation (Fig. 1).

a Thickness of insulation
b Price per board foot of insulation
c Fixed charge, per cent, including interest, taxes, maintenance and depreciation.

Cost of Refrigeration or Heating (Fig. 2).

d Conductivity and economic thickness of the material whose economical thickness is to be determined
e Conductivity and thickness of other parts of structure, such as structural wood, etc.
f Resistance of air spaces and surface resistance
g Difference between the temperatures of the air on the two sides
h Percentage of year in which the refrigerant or heat is employed
i Price of ice, refrigerant, or heat delivered to room to be insulated.

Transportation. For railroad-owned refrigerator cars a transportation charge is to be included (Fig. 3).

j Thickness of insulation
k Density of insulation
l Miles per year
m Cost of transporting extra weight of insulation.

These factors are discussed at the close of the paper. Their application to practical problems is brought out by a study of the accompanying charts.

Graphical Charts

6 Fig. 1, which has to do with fixed charges on insulation, is one of the simplest forms of alignment charts and may be used for calculating the yearly investment as related to interest, taxes, maintenance, and depreciation. The positions of the names of insulating materials on the chart are to be used with due caution, for the prices as indicated by the positions of the arrows may vary considerably, depending in part on the amount purchased,

the grade of insulation, and the market prices. For most materials it is safer to be guided by the actual quotations from manufacturers. Suppose the question is to find the annual charge per square foot for 2-in. corkboard selling at a uniform unit price of 6 cents per board foot for all thicknesses and with a fixed charge of 13 per cent per annum. Solution: Since the cost of insulation per board foot is 6 cents the "first cost" for 2-in.

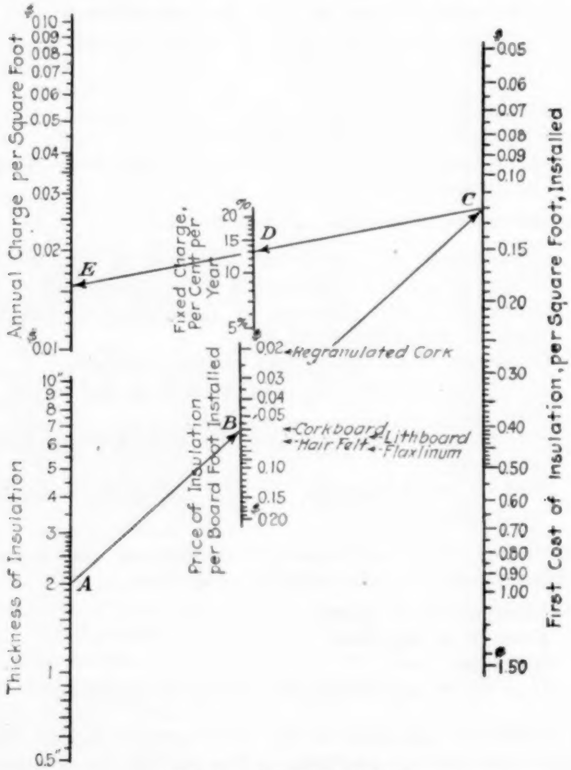


FIG. 1 INVESTMENT COST

insulation will be $6 \times 2 = 12$ cents;[1] which with a fixed charge of 13 per cent a year will give $12 \times 0.13 = 1.56$ cents or \$0.0156 per sq. ft. This answer may be read at a glance by following the straight lines *ABC* and *CDE* of the alignment chart,[2] Fig. 1,

[1] The line *ABC*, Fig. 1, is to be used only when the price per board foot is directly proportional to the thickness.

[2] For description of method of preparing alignment charts see chap. iii, Graphical and Mechanical Computations, by Joseph Lipka. The principles underlying the construction of nomographic or alignment charts have been fully developed by M. d'Ocagne of Paris in his Traité de Nomographie.

starting with 2-in. thickness of insulation. The scales on these charts should cover the usual range of values for the class of problems encountered, and the results for any particular case may be read directly and variations of individual values noted. This is essentially a multiplication chart which covers a wide range of values for the factors involved.

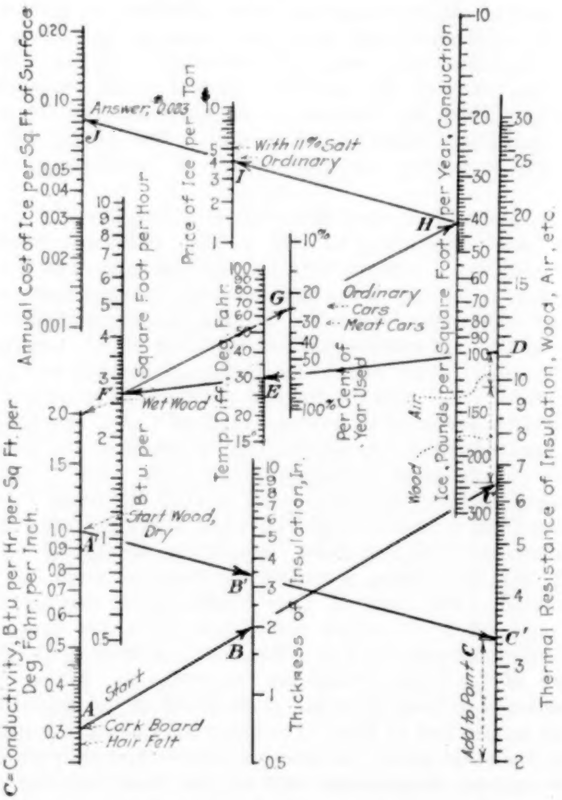


FIG. 2 ANNUAL COST OF ICE OR REFRIGERATION

Average values for surfaces as used in above chart: One surface, resistance, 0.8; two surfaces, still-air resistance, 1.6; one surface, air moving at 50 ft. per min., 0.5; one surface, air moving at 1000 ft. per min., 0.3. For air spaces, use reciprocal of heat-transmission values as obtained from Bureau of Standards tables. (See Kent's M. E. Handbook, 10th edition, p. 629).

7 Charts of similar character may be used for many classes of problems in which all but one of the variables are known or assumed. If several materials are to be compared, the same procedure, in general, can be used to get the annual costs for various thicknesses.

8 Fig. 2 is a chart constructed primarily for calculating the annual cost of ice, but may be used for the cost of refrigeration or heat. For very accurate work the engineer should carefully determine the conductivities of the materials to be studied, using values given by trustworthy authorities for the density, moisture content, structural condition due to years of service, etc. Data on some of the more common dry materials for various compositions and of different densities were published in 1915 by the Bureau of Standards and have been reprinted in various books and articles since they were first published.[1] On Fig. 2 will be found the names of two materials adjacent to the values of conductivities which are believed to be approximately correct for these insulating materials when new. The prices will vary according to market conditions, grade of insulation purchased, etc.

9 A sample floor calculation will illustrate how the graph is to be used. Starting at *A*, Fig. 2, with corkboard, conductivity = 0.308, lay a straight edge through the same trial thickness as in Fig. 1, namely, 2 in., and read on the right-hand scale a thermal resistance of 6.5. To this add the inside surface resistance of 0.8 and outside surface resistance of 0.5 (assuming a velocity of 500 ft. per min. as an average of standing and running while iced). Also, add the thermal resistance of structural wood obtained by assuming an average value of $C = 1$ and projecting a straight edge through 3⅜ in. (the thickness of the structural wood), which gives a resistance of 3.37. The total floor resistance is $6.5 + 0.8 + 0.5 + 3.37 = 11.17$ at point *D*. Now starting from a resistance of 11.17, lay a straight edge through "Temperature Diff. Deg. Fahr.," say, 30 deg. and get 2.7 B.t.u. per sq. ft. per hr. Again, reversing the direction from 2.7 B.t.u., go through "Per Cent of Year Used," taken here as 25 per cent, and get 41 lb. of ice melted per sq. ft. per year due to conduction through the floor. Reversing, go from 41 lb. through "Price of Ice" (or refrigerant or heat), say, \$4 per ton of ice delivered to bunkers of car, gives \$0.083 as the annual cost of ice per square foot of floor. For other thicknesses it is necessary to use the chart again; the answers are not directly proportional to thicknesses, because the sum of the wood and the surface resistances does not vary with change in thickness of the insulating material.

10 The scales of Fig. 2 were chosen particularly for refrigerator-car insulation. This same chart, when used within the limitations of the scales, may be used in solving problems where the refrigeration is supplied to the cold-storage room by a circulating medium. In this case the upper right-hand scale may be changed from pounds of ice into B.t.u. by multiplying the figures on the chart by 144 and marking the divisions as B.t.u. The middle upper scale of Fig. 2 may be made to represent the

[1] **Kent's M.E. Handbook, 10th edition, p. 630.**

cost of heat per B.t.u. transferred from the refrigerating room, but the conversion of this scale from the basis on which it is now given on the chart to one showing cost per B.t.u., becomes a problem involving many factors under two items, (*a*) capital costs, (*b*) production costs, and is outside the range of the present discussion.

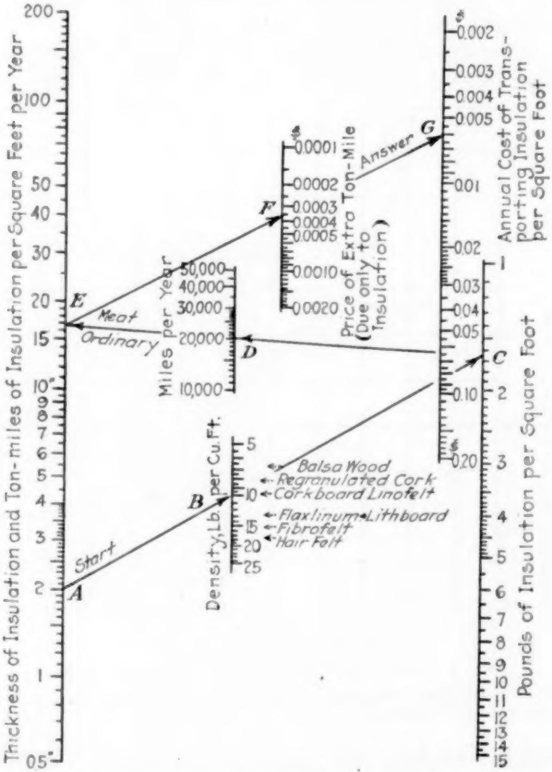


FIG. 3 ANNUAL COST OF TRANSPORTING INSULATION

11 For certain problems the range of these graphs may not suffice, but the scale may be extended by logarithmic or other mathematical relations. Where many factors are involved this alignment type of chart is found superior in accuracy and compactness to the more conventional rectangular-coördinate type originally prepared and used by the authors for insulation work.[1]

12 If transportation of tare weight of cars concerns the owner of the car, as in the case of cars owned by railroad companies,

[1] Referred to by the authors in their discussion of paper on Economic Thickness of Insulation, by P. Nicholls, December, 1922, meeting A.S.M.E. and A.S.R.E. See *Refrigerating Engineering*, Feb. 1923.

the charge should be added to the sum of ice and investment charges.

13 The amount of transportation charge for extra weight of insulation can be determined from Fig. 3, the procedure being similar to that for the other charts. One should not depend on the densities of materials shown, but should determine the actual densities by weighing samples from stock. For example, balsa wood and hair felt may vary widely in density.

14 The prices of transporting extra weight of insulation cannot be accurately determined, but the authors would submit

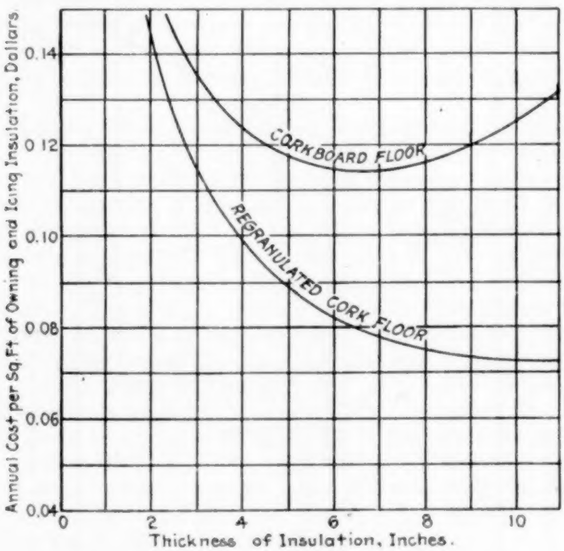


Fig. 4 Insulation Economics for Meat Cars

Based on 35 deg. fahr. temperature difference; 3.3 in. of dry wood in the floor; 11 per cent salt used with ice.

\$0.000625 per ton-mile of extra weight on level routes, \$0.00305 for mountain roads, and \$0.00108 for transcontinental service as being the best approximations available as brought out by them in a discussion of the cost of transporting extra weight of cars, which will form a chapter of a bulletin to be issued later. For purposes of illustration, the value of \$0.00035 has been assumed (point *F*). The final answer is at *G* and gives 0.6 cent per sq. ft. as the annual cost of transporting the insulation in the above example. In Tables 1, 2, and 3 and in Figs. 4 and 5, the more nearly average value of \$0.00108 was used. The significance of the weight factor is brought out by changing the position of line *EFG*, Fig. 3, to cause it to pass through some other point on the middle scale.

Economic Curves and Results

15 One of many applications of the foregoing may be indicated. Having thus calculated all the charges for various thicknesses of various insulating materials, economic curves similar to Fig. 4 can be plotted for floors under prescribed conditions of operation. Other curves can be drawn for ceilings with their own peculiar conditions. Fig. 5 shows three economic curves for ceilings and floors of refrigerator cars engaged in general perishable-fruit business, the data for plotting the curves being taken from Tables 1, 2, and 3 where the values given are in dollars

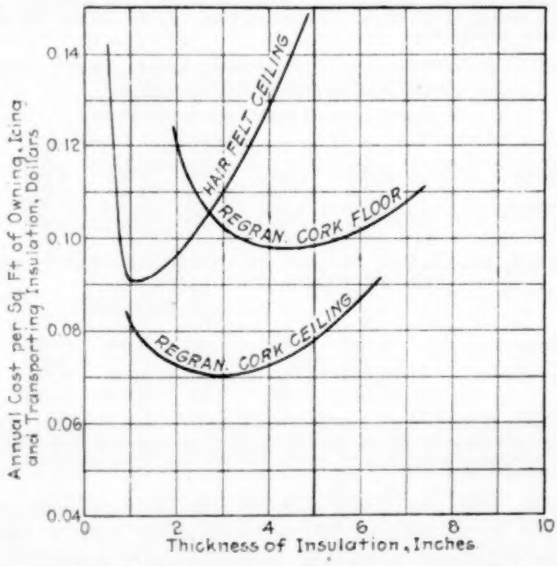


Fig. 5 Insulation Economics for Transcontinental Fruit Cars

(See Tables 1, 2, and 3 for details.)

per sq. ft. of surface per year. The condition under which standard fruit cars operate in average transcontinental trips from April to September is shown in the notes under the tables.

16 A comparison of the curves in Fig. 4 reveals the economic superiority of regranulated cork for floors. The curves show that the minimum annual cost per square foot of corkboard is $0.114, whereas with regranulated cork the minimum is $0.072. The respective economic thicknesses are seen to be 6.5 in. and 10.5 in. If the structural wood were considered to be wet, and this is the safe assumption, the relative advantage of regranulated cork as compared with corkboard would be even greater. If steel floors were used instead of wood, regranulated cork would also show a greater economic advantage.

17 Fig. 5 shows that even for the same material the economic thickness may be much more for floors than for ceilings. Probably the economic thickness for side insulation would be different from roofs or floors because the average temperature difference in the side is different from other parts of the car and the structural wood and number of air spaces may be different. The curve for hair felt[1] appears to be a minimum at 1 in., but in practice such a thin ceiling should not be used in the car because of the greater tendency for the top layers of fruit to spoil. The economical thickness of ceiling insulation for overhead refrigerating systems, such as the A. B. C., Moore, or the side-draft bunker system,[2] would be considerably greater than the ceiling thickness shown in Fig. 5, because of the superior refrigeration in the upper part of the car. On the other hand, the floor thickness could be reduced in the above-mentioned cars because in such cars the floor temperature need not be carried so low to obtain a good average temperature throughout the load.

TABLE 1 ECONOMICS OF TRANSCONTINENTAL FRUIT CARS — HAIR-FELT CEILINGS *

	Thickness in inches					
	1	2	3	4	5	6
Fixed charges	0.009	0.018	0.027	0.036	0.045	0.054
Ice	0.067	0.047	0.038	0.031	0.026	0.023
Transportation	0.015	0.031	0.045	0.063	0.079	0.095
Total	0.091	0.096	0.110	0.130	0.150	0.172

* Based on service from April to September, inclusive. Average shade temperature, 64 deg. fahr.; fair-weather factor, 70 per cent; average increment on roof, 8.2 deg. × 70 per cent = 6 deg.; average total roof temperature, 64 + 6 = 70 deg.; average ceiling temperature (2 in. below), 50 deg.; structure wood in ceiling (2.5 in.), dry; air spaces in ceiling, 2 in.

TABLE 2 ECONOMICS OF TRANSCONTINENTAL FRUIT CARS — REGRANULATED-CORK CEILINGS

(Footnote to Table 1 applies also to this table)

	Thickness in inches					
	1	2	3	4	5	6
Fixed charges	0.003	0.006	0.009	0.011	0.014	0.017
Ice	0.070	0.051	0.040	0.033	0.029	0.025
Transportation	0.008	0.015	0.022	0.029	0.036	0.044
Total	0.081	0.072	0.071	0.073	0.079	0.086

TABLE 3 ECONOMICS OF TRANSCONTINENTAL FRUIT CARS — REGRANULATED-CORK FLOORS **

	Thickness in inches					
	2	3	4	5	6	7
Fixed charges	0.006	0.009	0.011	0.014	0.017	0.020
Ice	0.100	0.071	0.060	0.047	0.042	0.037
Transportation	0.015	0.022	0.029	0.036	0.044	0.051
Total	0.121	0.102	0.100	0.097	0.103	0.108

** Based on service from April to September, inclusive. Average shade temperature, 64 deg. fahr.; average increment under floor, 3 deg.; average total floor temperature, 67 deg.; average temperature 2 in. above floor, 37 deg.; temperature difference, 30 deg.; structural wood (3¾ in.), dry.

[1] As brought out in the discussion, Keystone hair felt was used to illustrate the particular case under consideration and its conductivity and density were taken from tests made in 1916 by the Bureau of Standards.

[2] See *Railway Mechanical Engineer*, June, 1923, or *Refrigerating Engineering*, June, 1923, Tests on a Refrigerator-Car Model, by P. X. Rice.

18 A comparison between the curves of Figs. 4 and 5 indicates a great difference in economic thickness. This is due to the lower inside temperatures of the meat car and to the fact that meat packers do not have to pay for transportation of extra weight of insulation. With the present system of car interchange, the cost of transporting the extra weight of a thoroughly insulated car is not assessed against the car owner, even if the owner is a railroad company. If only one road should adopt thicker insulation, the other railroads, over whose lines the car operates, would unwittingly bear the extra cost of transporting extra insulation. Naturally, this loss would come out of what would perhaps otherwise be profit for the roads handling the car. Therefore, as interchange customs now stand, the economical thickness of insulation should be calculated with practically no transportation charge. To be exact, of the car's annual mileage of, say, 20,000, the owner of the car may haul it only 3000 or 5000 miles, and the transportation factor should be reduced accordingly. Small roads owning refrigerator cars would therefore insulate them more thoroughly than would long roads. In fact, the short road might well neglect the transportation cost of thick insulation and use insulation much thicker than is now commonly employed. Furthermore real thick insulation would reduce spoilage claims somewhat and obviate much criticism from shippers.

Air-Space Construction

19 Cellular paper insulation has possibilities in connection with the economics of refrigerator cars, principally because of its light weight. The natural criticism to such a suggestion is that paper insulation would disintegrate in the presence of the inevitable moisture that accompanies refrigeration. The authors do not know of any tests on such material, except those made at Pennsylvania State College on a piece of paper insulation 1 ft. square and 1 in. thick, with one edge nailed to an improvised flag mast. This insulation has withstood the rain, snow, sun, and gales for one year, and apparently would still function as refrigerator-car insulation. It was constructed of hard glazed wrapping paper, coated on both sides with shellac to fasten together the corrugated sheets and to waterproof the paper. Half-inch corrugations were put between flat sheets of paper, and two of such series constituted the 1-in. thickness. The authors also constructed paper insulation 3 ft. square and 2 in. thick in a similar arrangement. This weighs only one-eighth as much as hair felt of equal thickness or one-quarter as much as hair felt of equivalent thermal resistance.

20 Objection has been made to air-space construction on account of the uncertainty of the air spaces remaining securely sealed. Apparently no dependable quantitative data on air

leakage in air spaces are available, but information from a number of engineers who have examined the conditions after cars have been in service, leads the authors to believe that in most cases air spaces are maintained for a long time without any serious change. The " heat meters " which are being developed independently by the Engineering Experiment Station and by the Research Laboratory of the American Society of Heating and Ventilating Engineers, may be used to measure directly the effectiveness of old and new cars of various constructions. Until satisfactory tests are made to settle the difference of opinion on the question of air leakage, the authors will continue to believe that there are undeveloped possibilities in air-space construction.

Discussion of Factors Involved

21 There are not many factors involved other than those named which affect the choice of insulation for best economy. Moreover, most of those listed are subject to variation in characteristics difficult to evaluate. Further consideration may be given to the factors mentioned early in the article and these will be taken up in the order as listed.

a Thickness of insulation may not conform to exact specified thickness. This is particularly true of hair felt.

b Price per *actual* board foot may not conform to *quoted* price on account of discrepancy in thickness.

c Fixed charge, in per cent, includes maintenance and depreciation which change with successive improvements in construction and operation, and is somewhat difficult to determine accurately for any particular installation.

d The conductivities of most of the insulating materials have been determined for certain conditions, but unless the engineer knows something of the properties and characteristics of materials and the conditions under which they were tested, he may be misled in using tabular values.

The conductivity of a given piece of insulation is almost certain to lose some of its thermal resistance in service. During the process of installation the material may be crushed or stretched or fail to fit closely enough to prevent convection through the wall or part of the wall. In service, the wall insulation may become detached in places and break the continuity of the material. From an inspection of hundreds of cars the authors concluded that usually car floors and the lower part of walls are soaked with water. Superintendents of refrigerator service admit that floors and lower walls are the least satisfactory. Unless steel floors are used, it would be safer to assume that the floor insulation will be wet and the conductivity should be chosen accordingly. In the case of yellow pine, the authors found that the conductivity of wet wood was about twice as much as that of

dry wood. With insulating materials the variations due to moisture content are not known and the true values cannot be used until some laboratory has determined the effect of moisture. For imperfectly protected floor insulation the authors suggest that conductivity be doubled for economic calculations. A certain railroad built two milk cars with watertight, rust-resisting steel floors; probably the reduced maintenance cost would justify such construction independently of the considerable saving in the cost of ice.

e In general, the conductivities of other structural parts of a car cannot be determined accurately, but ordinarily the thermal resistance of such parts is such a small part of the entire wall or floor that some tolerance of conductivity is permissible. As

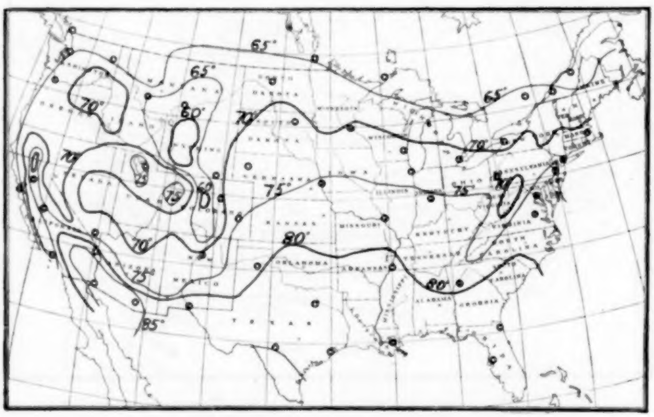


FIG. 6 AVERAGE JULY SHADE TEMPERATURE

mentioned before, the thermal resistance of wood depends largely on its moisture content and probably, for unprotected floors, should be taken as half the resistance of dry wood. Other structural materials often have less insulation value than is shown in tables for conductivity.

f Resistance of an air space is difficult to predict unless it is known to be strictly airtight. For stationary cold-storage walls, ordinary cracks would not be as objectionable as in moving cars. It has not been determined how tight an air space must be in order to be effective. Surface resistance is such a variable that the safe procedure is to use a value of zero for the outside surface of cars when the air passes over the surface at a speed of more than 10 miles per hour. If the resistance of the car wall proper is relatively high, as in modern refrigerator cars, the outside surface resistance can be neglected with but slight error in the total transmission through the wall. It is better to use some

approximation of the outside surface resistance, taking into account the velocity of air on the surface. The inside surface resistance amounts to more than that of the outside, and ordinarily where the air is fairly calm, the surface resistance can be predicted closely enough for thick walls where an error in the the surface resistance would not affect seriously the total transmission.[1]

g Differences of temperature between the outside and the inside of the car should be determined by actual measurements; if that is not possible, it can be approximated by aid of recorded tests of similar cars, provided it is known where the temperatures have been observed. The following may assist in this approxima-

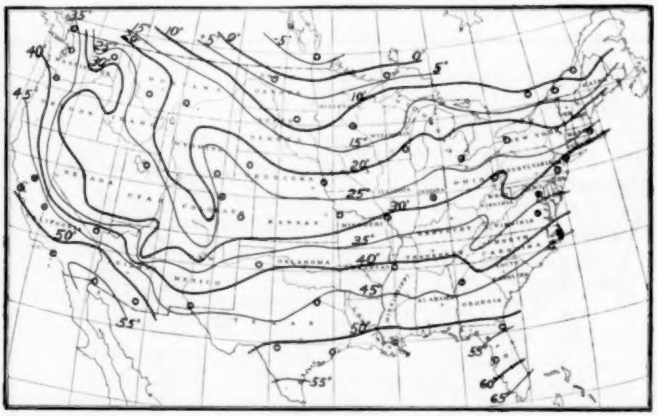


FIG. 7 AVERAGE JANUARY SHADE TEMPERATURE

tion: The floors of end-bunker cars are often 20 deg. to 35 deg. colder than the ceilings. It is desirable to use the temperatures taken at least 2 in. from the car surface and to include surface resistances in the calculations.

Outside shade temperatures can be predetermined approximately by the temperature maps, Figs. 6, 7, and 8, which show July, January, and annual shade temperatures. These apply to outside wall and floor temperatures, but are somewhat too low because the walls receive direct radiation from the sun on bright days and floors receive some radiation from hot ballast. In the case of roofs, especially black or red roofs, the error of neglecting radiant heat is appreciable, as is brought out in Fig. 9. The authors have found that at noon, summer or winter, at any latitude, there is a difference of about 62 deg. between shade and

[1] This is further discussed in an article by A. J. Wood on Insulation of Passenger and Refrigerator Cars, in *Railway Mechanical Engineer*, September, 1921.

either black- or red-roof temperatures,[1] provided the sky is clear and the roof is perpendicular to the sun's rays. If the roof is in a horizontal position, the variation in temperature increment above shade temperature is shown in Fig. 9 by the broken lines for August, October, and December. For an integrated average covering a period of 24 hours' fair weather, the increments obtained are 20.4 deg. in August and 17.5 deg. in October, or 17 deg. annual. The corresponding increments for white roofs are only 7.6 deg., 5.6 deg., and 5.5 deg. It can be seen that if the railroads would paint their car roofs white, as some paint the walls, considerably less ice would be required. However, the 20.4-deg. average for August for black roofs applies only for still air in clear weather. Refrigerator cars stand still only about

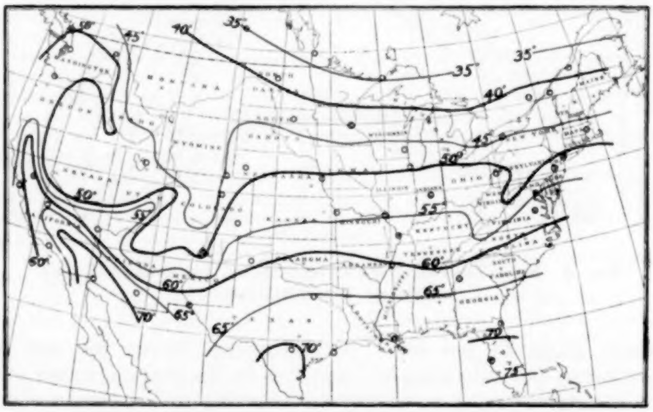


FIG. 8 AVERAGE ANNUAL SHADE TEMPERATURE

40 per cent of the time they carry ice, and much of the other 60 per cent they run at speeds above 10 m.p.h., above which speed the temperature increment is negligible. Furthermore the natural wind velocity is often greater than 10 m.p.h., so that one should not expect to experience more than 40 per cent of 20 deg. = 8.2 deg. average effective increment. For stationary cold-storage buildings the effective increment in August would probably be 15 deg. to 18 deg.

By aid of the diagrams of Figs. 6 to 9, inclusive, one is able to reach fairly accurate results for outside average-temperature conditions in different parts of the country and for different conditions under which the sun strikes the surface. Inside temperatures may be approximated as indicated, making it possible to determine with reasonable accuracy average temperature differences.

[1] This value of temperature increment is approximate, being the average of many readings of mercury thermometers.

h The correct figure for the percentage of year in which a refrigerant or a heater is used should be determined on the car-day basis; that is, if only a few cars are refrigerated in winter and all the cars are refrigerated for three months in summer, the figure would be slightly over $3/12 = 25$ per cent. This is sufficiently close for ordinary refrigerator-car service, but for special industries, such as meat packing, the figure may be 30 per cent. In the case of cold storage the figure is generally much higher.

i The price of ice for refrigerator cars can be found readily by referring to the regular charges made at icing stations, ordi-

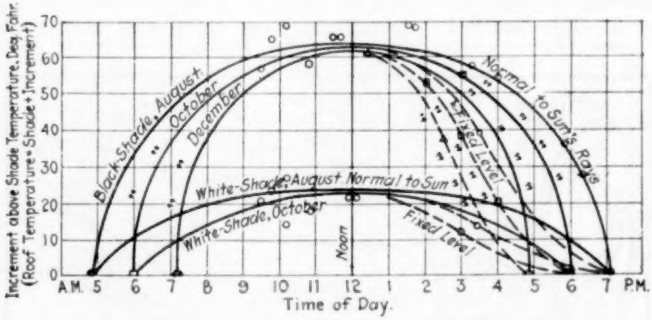


FIG. 9 ROOF-TEMPERATURE CORRECTIONS, INCREMENT ABOVE SHADE TEMPERATURE

narily about \$4 per ton. In cold-storage plants the cost of producing and delivering refrigeration to the storage rooms can be determined most accurately by referring to the construction, maintenance, and operating costs of the plant; probably the cost is less than \$1 per ton of ice (equivalent refrigeration). It may be noted that the cost of absorbing heat by refrigeration or by ice is many times more expensive than supplying that much heat by heaters. Therefore the cost of supplying heat to refrigerator cars in winter is almost negligible in comparison with the cost of refrigerating the same cars in summer; the economic thickness of insulation should therefore be determined by the refrigeration.

j See comment on thickness in paragraph (*a*) above.

k, l, and *m* have been covered by the discussion in previous paragraphs.

22 It should be apparent to those who have followed the discussion, that the most intricate problems of economic thickness of insulation may lend themselves to a satisfactory solution if full information is obtained concerning the factors involved.

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DISCUSSION

CHARLES H. HERTER. The authors are quite correct in stating that practically all refrigerator cars used in the United States are deficient in insulation. This new method of calculating the gain derived from more insulation should shortly lead to their improvement. It is not necessary to reserve these improvements for new cars only. Many old cars may have added to them one or more layers of insulation.

Not being influenced by manufacturers' literature, the authors seem to have discovered some merit in regranulated cork. Samples tested at the Bureau of Standards and elsewhere prove the heat conductivity of dry regranulated cork to be equal to that of pure corkboard, and yet the current prices per board foot, as indicated by Fig. 1, are in the ratio of six cents to two cents, while on the basis of 1-in. board equaling 2 in. of regranulated cork (manufacturers' practice), the latter would cost four cents. Consequently heretofore regranulated cork was always at a disadvantage, the makers preferring to sell materials in other than loose form. The objection that loose filling materials require lumber or the equivalent to confine them does not apply in the case of refrigerator cars, because to secure a durable wearing surface inside and out, wood sheathing is generally used, even with the self-sustaining corkboard. If forced shaking tests were made with various-sized granules of loose cork an idea would be obtained of the necessary mixture and density per cubic foot to secure the least amount of settling. Such settling is objectionable, at least in the case of vertical surfaces. In car walls the settling could probably be eliminated by mixing the regranulated cork with diatomaceous earth in powder form (Celite, Sil-O-Cel), the conductivity of which does not differ from cork. The weight averages 10 lb. per cu. ft., while the price is equivalent to about two cents per board foot. Celite is used for insulating furnaces because it is heat- and fire-resistive. But it should be kept dry, the same as all insulators. It is the only loose insulating powder which retains its normal density.

The authors justly call attention to the fact that care should be exercised in selecting the proper coefficient of heat transfer applying to the particular material in use. Materials are not likely to duplicate selected laboratory samples, and the least one should do is to compare the weights per cubic foot. Dense materials as a rule offer less resistance to heat flow than do light ones. In this connection the writer's collection of over 500 test results [1] attached to the 1923 report of the Insulation Committee of the American Society of Refrigerating Engineers should be of some assistance, as it contains also data on the effect of moisture.

It will be noticed, Par. 17, that the authors arrive at a temperature difference of 70 – 50 or 20 deg. fahr. only in the case of ceilings of fruit cars, against 67 – 37 or 30 deg. in the case of floors. Accordingly they show, Fig. 5, the economic thickness of regranulated cork at 3 and 4½ in., respectively. A refrigerating engineer would reverse these thicknesses, and place the thickest insulation on the ceiling to obtain better protection against radiation and convection, higher heat transfer of wetted roof area, and aim at securing a ceiling air temperature at the most 10 deg. higher than the air over the floor. It stands to reason that the thin ceilings of cars are largely responsible for the warm stratum of air under the ceiling. In other words, the temperature difference through the

[1] *Refrigerating Engineering*, vol. 10, no. 7, Jan. 1924, p. 256.

ceiling should be taken as 40 rather than 20 deg., especially over the ice bunkers. By elevating the roof line of a car to that of the locomotive, space would be gained for the extra thickness of insulation without affecting the cubic contents of the cars. To duplicate the refrigerating efficiency obtained in stationary installations, the ice or cooling surface should of course be spread over the car ceiling and not crowded to the remote ends.

The authors advocate air-space construction, even if made with several sheets of paper. This plan may be entirely feasible from the viewpoint of insulating effect only, but it is objectionable from the standpoint of refrigeration. The important factor of heat capacity of the car body (weight × specific heat) apparently did not receive its share of attention from the authors. Papered air spaces may insulate well per degree of difference, but having no weight, their own temperature will fluctuate with all temperature variations encountered by the car. Thus in precooling the car, air would be lowered to 35 deg. within a few minutes, but on the road the weightless walls would likewise become warm very soon instead of remaining "saturated with cold" for many hours. To keep down temperature fluctuations within the car as in a room the inner envelope should have capacity to hold a great deal of heat (or cold) and the insulation proper may form the outer envelope. In fact, it may then consist of air spaces; but weight is useful, especially if there is no tax imposed on tare weight.

Owing to their higher temperature, air spaces in the outer envelope are less effective than inside, but also more secure against condensation of moisture. When heat capacity is taken into account the opportunities of heavier insulators become brighter than would appear from the authors' deductions. A heavy lining of yellow pine, oak, or maple will be preferable to one of redwood. Light-weight woods belong on the outside. The exterior should be as smooth as possible and as stated by the authors should be painted all over with glossy white paint, both for appearance and to repel heat.

George A. Nicol, Jr. The authors open up a rather unique line of thought as regards the supposed economical thickness of insulation in refrigerator cars. So many variables enter into any such calculations that the elimination of any one factor, although apparently negligible, may upset the calculations and render theoretical results inapplicable to practical use.

In Fig. 1, the fixed charge per cent per annum has been used on the basis of 13 per cent. No explanation has been given as to how this figure has been ascertained. In Fig. 2 it will be noted on the scale of Ice, Pounds per Square Foot per Year Conduction, that this was a constant value, whereas it should be variable, since the amount of the ice required with a constant temperature difference would vary, depending upon the amount of salt used with the ice. For example, with the 11 per cent salt referred to

by the authors about 10 per cent more ice will be required for a given amount of refrigeration than if the ice is used without salt, because if salt is used the water will be discharged at a temperature of about 14 deg. lower than if no salt is used.

In Fig. 3 the first intermediate scale calls for the density in pounds per cubic foot, and opposite the various values on this scale are values for common insulating materials, 19 being given for hair felt. This figure is apparently for compressed hair felt, and as hair felt used for car insulation is not compressed, this value should be approximately 9. The value on the scale was undoubtedly taken from tables published by the Bureau of Standards, but does not represent correctly the density of the material in pounds per cubic foot. Accordingly this has led to miscalculation in the tables following which were used for drawing up the curves in Fig. 5.

The writer believes that the authors have used in the calculations figures obtained from the Bureau of Standards on Keystone hair felt, which is covered on both sides with a waterproofing paper. It is entirely different in construction from standard hair felt, although the term "hair felt" has been used throughout the paper.

The correct density of standard hair felt 1 in. thick is 9 lb. per cu. ft., and the conductivity in B.t.u. per sq. ft. per inch thickness per deg. fahr. difference in temperature per 24 hr. is 5.9, and for the Keystone hair felt the density is 10½ lb. per cu. ft. and the conductivity in B.t.u. per sq. ft. per inch thickness per deg. fahr. difference in temperature per 24 hr. is 6.5. These values, of course, will change materially the curve shown in Fig. 5 for the hair-felt ceiling. The most pronounced feature due to the lessening of the weight is the flattening of the curve for hair felt. Fig. 10 shows the insulation economics for both Keystone hair felt and standard hair felt. The third curve indicates the result obtained using the heat-conductivity value used in the paper for hair felt, but using the revised weight and interest values in accordance with the true figures for Keystone hair felt.

The economic thickness for the standard hair felt on the new chart is 2⅛ in., and for Keystone hair felt is 1⅞ in. These figures check very closely with recognized standard practice throughout the United States and Canada for refrigerator-car design, excepting that it has been found in practice to require not less than 2½ in. of insulation in the roof of either ½-in. Keystone hair felt or standard hair felt. This accords with the recommendations of the United States Department of Agriculture, and is based on results obtained from a great number of tests of transcontinental perishable-fruit trains. It is unquestionable that it is necessary to provide more and better insulation in the roof than in any other part of the car.

This brings in another variable that has been neglected, i.e., the loss and damage claims actually found in refrigerator-car operation due to spoilage of perishable products, especially the top layers as commonly carried in refrigerator equipment. Carriers report loss and damage claims amounting to $14,500,000 as paid in

1921 on fruits and vegetables alone. Fig. 5 does not, therefore, indicate the true economic thickness, since any reduction of insulation over present common and established practice in the weakest part of the insulation, that is, the roof of the car, would necessarily increase these claims considerably.

The paper throughout would seem to establish that a new field had been found for the use of regranulated cork, which consists of the sawings and trimmings from pure corkboard. Its one advantage of initial low first cost is offset by several factors. Its thermal value is 27.2 per cent less than standard hair felt and

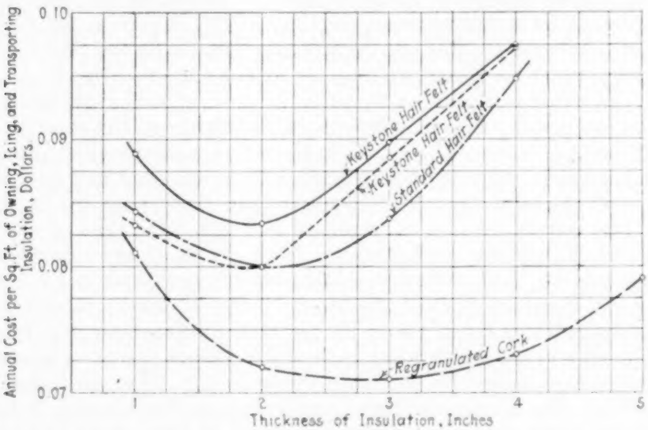

FIG. 10 CORRECTED CURVES FOR INSULATION ECONOMIES FOR TRANSCONTINENTAL FRUIT CARS

15.4 per cent less than Keystone hair felt per inch of thickness. Also, on the same basis of modern refrigerator-car equipment requiring $2\frac{1}{2}$ in. of insulation in the ceiling, it would be necessary to use not less than $3\frac{1}{2}$ in. of regranulated cork, with a consequent increase in weight per car. Furthermore, nothing is said of the practical application of regranulated cork, but it is assumed that this material is packed into place, and while the paper confines the calculations to the floor and ceiling, undoubtedly it would be the intention to use this same material in the side and end walls. This would present considerable difficulty as regards its application and permanency. The tendency of the material is naturally to pack, and due to the buffing shocks, vibration, etc., when refrigerator cars are in motion, it would tend, especially in the side walls, to leave an open space for a considerable distance between the side plate and the side sill of the car. It does not seem practical to allow any opening for applying additional regranulated cork as the mass settles.

Before regranulated cork is used for car insulation its absorption of water, which would materially decrease its power of insulation,

should be thoroughly considered. When outside air, which always contains some degree of moisture, is cooled due to leakage through the insulation, the degree of saturation is increased up to the dew-point. Condensation will then occur and water will be deposited in any granular filling material through which air may circulate. This moistening will begin with the interior portions of the material, and capillary attraction will tend to distribute the moisture throughout the thickness of the material. This tends very materially to increase the packing of the granular material and greatly lowers its insulating value.

Experience with cold-storage insulation has shown that granular materials such as sawdust, etc., become saturated with moisture throughout their mass. On the other hand, experience in refrigerator-car practice has likewise shown that hair-felt material, which is always applied in blanket form, is not seriously affected by moisture. As a matter of information, it is entirely possible today to reinforce hair felt as well as to waterproof it.

In the puncturing of side walls or periodical replacing of damaged outside sheathing there would be entailed a great loss of any loosely packed insulation such as regranulated cork.

The writer wishes to call attention to Dr. M. E. Pennington's discussion of W. H. Winterrowd's paper[1] before the A.S.R.E. and A.S.M.E., wherein she referred to the fact that air spaces in refrigerator cars are unsatisfactory for insulating purposes as it is impossible to keep the compartments tight. This has been proved in actual practice, and, while there is no question but that these spaces have a certain value, they do not compare in any way with refrigerator-car insulations now in common use.

P. Nicholls.[2] While fully supporting the free use of calculating charts, the writer questions the logic used in the second paragraph of the paper. Any chart must be plotted from an equation and is a sequence to it. As to whether they are needed or will be time savers depends on a number of factors, the two most important being as to how often and continuously they will be used, and the type of arithmetical computation involved. In this particular illustration only simple multiplication and division are involved, and could as speedily be done on a slide rule. It would have been an aid to have included an equation for the economic thickness derived directly from the factors.

It is also suggested that to obtain full advantage from such charts, where several are required for one calculation, and especially when they are to be used to plot curves such as Fig. 4, a typical ruled form should be included to show how the derived items should be recorded and summed up. Doing this gives the maximum protection to the user, and avoids the need of consulting the text each time they are used.

[1] *Refrigerating Engineering*, vol. 9, no. 2, Aug. 1922, p. 62.
[2] A.S.H. & V.E. Laboratory, Bureau of Mines, Pittsburgh, Pa.

As a large number of cars usually will be made to any one design adopted, an intense study probably would be warranted, and for each type being considered curves such as Figs. 4 and 5 would be plotted. It can also be believed that costs so derived would not be the sole deciding factor, since to only a few of the factors for or against each insulation and thickness could monetary values be assigned, and others would involve the probability of certain events happening. The final decision would then be by guess or authority.

The authors have done good work in collecting the data on the factors and showing how they are interconnected. If any further advance is to be made in the attempt to fix the thickness automatically by comparing monetary costs, it will be attained by tabulations of experience reduced to probability values.

H. J. Krampe. The purpose of the paper is commendable but the authors have apparently lost sight of the commercial factors in the preparation of it. These factors may be listed as the initial cost of material delivered; the cost of installation; and the depreciation, both in resistance to heat flow and in length of service.

In initial cost the items necessary to the proper erection or installation of the insulation must be considered in addition to the material itself. Car builders are at present giving this subject much attention, which is a probable reason for the recent tendency to change refrigerator-car design.

The writer agrees with the contentions of Mr. Nicol that the loss due to damage claims has been instrumental in improving the insulation of refrigerator cars, and that granulated cork and mineral wool pack. He does not agree with him that granulated cork will absorb moisture as rapidly as some of the insulators of a fibrous nature, having more or less capillary attraction, depending on its location in the car. The lower the position in the car the more necessary it is to eliminate capillary attraction and moisture conditions, which is probably the reason for the adoption of corkboard in the bottom of the car first. The Merchants Dispatch, which probably was the first to use corkboard in its cars, recently found that the corkboard installed in the bottom of the car eight years ago was in excellent condition. This substantiated their original conclusion that this was the proper material for the bottom of the car.

The suggestion that Celite might be combined with granulated cork, as a component to reduce packing, would be good were it not for the fact that Celite will absorb moisture and thus introduce a factor we are trying to eliminate, since moisture will depreciate the resistance to heat flow.

Air spaces will prove objectionable, for the reason that they mean a greater total thickness of car walls. They thus reduce the cubic contents of the car, and hence affect its carrying capacity.

Willis H. Carrier. The effect of moisture on the conductivity of insulation is a factor that should be emphasized. Most tests on insulation have failed to take this factor sufficiently into account. The test pieces have not corresponded to actual conditions, where there is leakage and infiltration causing much precipitation of moisture.

Any porous insulating material, particularly if it is hygroscopic, will become practically saturated when used with a cold surface on one side and air of high moisture content on the other. Tests for insulating values should be made under such practical conditions, and there would probably be a wide difference in values if this were done. Our experience in insulating air ducts and piping is that nothing is equal to corkboard. Although it is more expensive in unit cost than other materials, the actual expense of the other materials is greater because they do not give as good results. They require a greater thickness to be used, largely because of this factor of moistening in an atmosphere that is of higher dewpoint than the temperature on the interior.

Fred Matthews. The paper under discussion should be taken as *evidence* of what may be expected under certain conditions and not as *proof* of what will actually occur under other conditions.

There are always certain dangers inherent to preliminary discussion regarding things that have not been reduced to practice, due to the fact that the results given, in such a paper as this, for example, may be misinterpreted, the evidence being accepted as law. To illustrate, reference to paper, either corrugated or plain, may lead to attempts at the actual building up of insulation from many thicknesses of paper, which would have all of the disadvantages of air spaces, and in addition the high capillarity of the several sections of paper. In low-temperature work the air enters saturated with moisture, and on encountering the lower temperature deposits its moisture and passes out, the process being repeated until the insulation becomes saturated or, in case of very low temperatures, the air spaces become filled with ice.

Regranulated cork, which is sheet cork, has a high insulating value and as such is valuable. The principal difficulty regarding regranulated cork is to keep it contained! Where regranulated cork is packed in a wall the slightest air leak, if there is an outlet as well as an inlet, may actually sweep this fine material out; and if a leak occurs at the bottom of the space filled it will actually run out like sand through an hour-glass. Regranulated cork is not readily water-absorbent as has been implied and should not be compared to "sawdust."

The Authors. For the most part, the discussion has touched upon the less significant features of the paper. The authors are interested only in presenting a method for solving a certain class of problems and the values taken were primarily to illustrate the method.

In attempting a satisfactory solution, it was necessary to study carefully into new fields, where the data were either meager or were unclassified. The following are among these questions: (*a*) The extra cost to haul extra weight of insulation and the bearing of this factor on economical thickness; (*b*) method of approximating the average temperature differences between inside and outside of a car. Properly applying Fig. 6, 7, or 8 may prove of considerable service to designers; (*c*) roof-temperature corrections as brought out in Fig. 9 give the results of an individual study of a neglected factor, where radiant heat is important.

Mr. Nicol implies that the elimination of any one factor may upset the calculations. The authors fully appreciated this and in closing the paper stated that the most intricate problems may be solved "if full information is obtained concerning the factors involved." The most elusive factor is loss and damage claims. It does not seem desirable even to attempt to introduce factors which depend on so many variables beyond the control of the shipper, since in the problem under consideration we attempt to base the costs only on figures which would not vary widely for different shipments. Even with perfect insulation, railroads may be liable for spoilage claims, partly on account of the present design of bunker systems, over-ripe fruit, delays, flat wheels, etc.

Mr. Nicol is right in stating that the upper curve, Fig. 5, was for Keystone hair felt and that the values for conductivity and density were taken from the tables by the Bureau of Standards. We were aware that this brand of hair felt was frequently used in refrigerator cars.[1] Our curves in Fig. 5 are correct for the values assumed and his curves, Fig. 10, are also correct for the conditions he has taken. They both bring out the importance of the factor of weight, a consideration generally overlooked. The authors have recently learned that the Bureau of Standards' tests on "hair felt" and "Keystone hair felt" were made in 1916, and it would be impossible now to check the density of the particular samples tested. The density figures given for these two materials are considerably higher than should be expected for these materials as they are actually used at the present time. The density was originally measured by weighing the samples and dividing by the volume when actually in place in the test apparatus. Compression would therefore increase the density.

Mr. Herter adds some valuable points. We question his suggestion of making ceiling temperature only 10 deg. warmer than floor temperature. Tests do not bear him out, except in case of the less common overhead system of bunkers such as A. B. C., Moore, etc. The average trip of a refrigerator car is only for two or three days and during this time the difference between floor and ceiling in end-bunker-type cars was found to be what we have shown. Furthermore, with the end-bunker cars a considerable temperature

[1] Note the tabulation in Mr. Winterrowd's paper, Trans. A. S. M. E., vol. 44, p. 163.

head is required to circulate air, and although thicker ceilings might reduce ceiling temperature somewhat it would not be as much as might be expected, because the reduced circulation would not be able to keep the ceiling at a much lower temperature.

The specific heat capacity of insulations probably would enter into cars insulated insufficiently or only moderately; however, this does not seem to be serious in amount, and would be less for cars with generous insulation. The structural walls of the car may afford more specific heat capacity than the insulating materials.

Mr. Krampe evidently overlooked the point that the paper is devoted exclusively to the "commercial factor"; economics is nothing if not commercial. The factors he mentions as omitted are there, and others can readily be introduced if accompanied by sufficient and authentic data. We would also state that between the roof and ceiling and between the floor and bottom of the underframing there is ordinarily room for about ten inches of insulation without reducing cargo capacity, but the wall space is limited. Opinions seem to conflict on the question of keeping air-space compartments tight. The best information we have is favorable, at least up to two or three years of service. For a longer period, we have not been able to secure reliable information or data.

We are fully aware of the difficulties of keeping regranulated cork contained, as noted by Mr. Matthews. On the other hand, where the economic value is so evident, the practical difficulties should not prevent further efforts to meet this apparent shortage.

The authors do not feel the same concern as does Mr. Matthews for fear that the results will be misinterpreted. Those who fail to grasp the meaning of a paper of this kind may distort the facts, but those who are concerned with the analysis of the problem and the design of insulating structures should not be misled. In his first sentence, Mr. Matthews sums up the intent of the authors.

It is true, as intimated by Mr. Carrier, that the effect of moisture on the insulating value of materials should not be lost sight of. This is a problem which should bear careful investigation by use of one of our new "heat meters," referred to in Par. 20 of the paper.

No. 1910

THE FUNDAMENTAL PROBLEMS OF HYDROELECTRIC DEVELOPMENT

By JOHN R. FREEMAN[1], PROVIDENCE, R. I.
Member of the Society

WE are now in the midst of the greatest activity ever known in hydroelectric development and problems of unprecedented magnitude are before us.[2]

2 Coöperation between our great national engineering societies in discussing these great national problems, in which the works of the civil, the mechanical, and the electrical engineer are closely interwoven, is both pleasant and profitable. In the hydroelectric field the several engineering specialists must all work together from start to finish. One cannot begin where the other leaves off.

3 Today the most fundamental problems of hydroelectric development, interconnected power, and superpower are those of public relations and future public welfare.

4 This kind of a meeting may serve a useful purpose far beyond that of interchange of information between specialists in power development, if it can interest engineers working outside of this special field in the broad aspects of some of these fundamental problems.

5 Conferences on this topic will have their greatest usefulness if they can be made to arouse a campaign of education among the voters and their representatives in legislative halls, toward removing the present lack of understanding and make plain the true

[1] Cons. Engr. Past-President A. S. C. E., and A. S. M. E.

[2] We are told applications have been filed with the Federal Power Commission alone (which is far from covering all water-power prospects) for projects involving the installation of more than 21 million horsepower and that they have issued permits and licenses for an aggregate of 7 million horsepower. Meanwhile the sum total of water horsepower already developed in the U.S. is only 9 millions. (Figures on authority of O. C. Merrill in the address mentioned in following footnote.)

Perhaps, like most water-right filings in the West, these power-right filings are too optimistic in their estimates; nevertheless these figures tell a story of a wonderful awakening.

Address at the Hydroelectric Session of the Annual Meeting of The AMERICAN SOCIETY OF MECHANICAL ENGINEERS, New York, December 3 to 6, 1923.

relation to public welfare of widespread transmission of hydroelectric power.[1]

Scope of Discussion

6 The committee on this joint meeting has asked me to open this evening's discussion.

7 The A.S.M.E. Secretary tells me that society publications have been overfed with papers on station detail and that it is desired we now confine our talk to fundamentals.

8 I am asked to round up and set forth, so far as I can, briefly, some of the chief underlying principles. Others who follow will amplify my remarks so far as the brief time allotted to each speaker will permit.

9 None of us in this time and place can treat his subject exhaustively or minutely. The literature upon the details of power development would go far toward filling the famous five-foot bookshelf, and there is much of interest in recent developments that remains to be written up.

10 Agreement is not to be expected among all of us, coming, as we do, from different fields, with different points of view; nor can we settle the great problems here. Our chief purpose must be to promote and direct discussions along useful lines among engineers, business men, and those in public life.

11 The very fundamentals can be quickly stated:

1 A hydroelectric prospect must show a profit on the investment.

2 It should be so developed as to contribute to public welfare in the highest degree.

12 These principles seem so self-evident as to require no extended discussion, but to prove up, on their application to a particular prospect, may require an enormous amount of work.

Change and Progress within Speaker's Memory

13 Possibly the committee selected me to open this discussion in the expectation that I would say something in a sketchy way of the fundamental changes in these problems that have come about within my memory, and compare past conditions and the methods of solving their problems with present conditions, as a preliminary to suggesting the directions in which to look out toward future progress from present achievement.

[1] This matter of creating a better understanding by the public in hydroelectric development and of the influence of hydroelectric development on the welfare of the state was admirably set forth a few months ago in address by Mr. O. C. Merrill, secretary of the Federal Power Commission, before the Empire State Gas and Electric Association, reprinted in the *Electrical World* of Oct. 20, 1923; which issue also contains a tabular view of the location in different parts of the U.S. of generating capacity (steam and water power), aggregating 17,715,484 kva., figures that the mind can hardly grasp.

14 It is a little more than 50 years since I began work in the engineering office of the water-power company at Lawrence, Mass. At that time Lowell, Lawrence, Manchester, and Holyoke had come into their prime, and often were visited by engineers from other parts of the United States and from foreign lands; and often, as a youngster whose time was not particularly valuable to his employer, it was my good fortune to show these visitors around.

15 Fifty years ago, there were no better water-power developments than these, in America, or in the world. Here at Lowell and Lawrence the great problems of turbine versus overshot or high-breast water wheel had been worked out by Boyden, Francis, and Swain, and here turbine efficiencies had been achieved which stood unchallenged for nearly half a century. And in the testing flume (or hydraulic laboratory) at Holyoke the way had been shown toward obtaining larger power in smaller compass than had before been even dreamed of. Also at Lowell and Lawrence, in these early days, under Storrow, Francis, and Mills, the art of measuring large quantities of flow of water accurately had been worked out better than elsewhere. Storrow was the best-educated engineer of his day in America. Boyden, Francis, Swain, and Mills were men of genius.

Costs: Hydro vs. Steam

16 The fundamental question about water power that I heard most debated 40 to 50 years ago was its relative cost compared with steam power. That question is still with us. Every customer of the power company had ideas on this topic; but we had object lessons that water was cheaper in the price that the water-power company actually obtained from those factories which had emergency steam engines standing idle.

17 Rates for water at Lawrence had been shrewdly put at nearly as much as the traffic would bear, about $20 per hp. per year for the mere right to draw water for power from the canals nominally 16 hours per working day but actually about 10 hours. At Holyoke the price had been much smaller, but all primary power had been sold out long ago, mostly to paper mills for 24-hour use.

18 The earlier American factories nearly all sought water-power sites, but the perfection of the steam engine with automatic cut-off already had brought it into close competition with the water wheel. Power cost is but one of many factors determining location, and more steam-driven cotton mills were being built 40 and 50 years ago than those driven by water power. It was found that prompt and cheap transportation for raw stock and finished product, or the easy labor market of great cities, were often stronger attractions than a small saving in power cost for textile mills and machine works, where power was a small proportion of total cost of manufacture. Moreover, steam power often could be made cheaper than water power for woolen mills, dye houses, etc., by conserving the heat of the engine exhaust.

19 Steam power presents to the capitalist one great advantage in the degree of certainty with which a preliminary estimate of cost of complete plant can be made. The margin for doubt, accident, and contingency is small with steam, while the opposite is true with regard to water-power development because of dangers of a destructive flood that may overwhelm cofferdams, bad foundation conditions for dam or wheel pit — revealed only after excavation has been largely opened up — and sometimes grievously large awards for flowage damage.

20 Most of the southern New England water-power sites that could be developed economically had been already occupied, and many water powers on streams of small and irregular flow had been developed that would not today be worth while; although once built and the cost absorbed, it still pays to run them. Where water power had fixed the location, often growth in power demand had to be met by steam, regardless of cost, and a common condition found at the larger country mills was that of employing more steam power than water power. Lowell, 40 years ago, had come to have twice as much steam power as water power (as I remember the figures).

21 Fifty years have brought a wonderful increase in the amount of power used and in methods of developing it, but costs per horsepower in the broad view show small change. The cost of coal has doubled, but higher steam pressures, larger power units, the steam turbine, and the economies of the central station have cut coal consumption per horsepower-hour in half.[1]

22 In water-power development the unit costs per pound for machinery and per cubic yard for structural material have doubled. Although the efficiency of the hydro unit has increased at best only from 83 per cent to 93 per cent (and for some time has been close to the end of the road), the costs of installation, maintenance, and supervision have been held down by a wonderful concentration of power into single units, under terrific head, and by a revolution in the features of design for large developments, so that with all these changes there has been remarkably small change in 50 years in cost of either hydro or steam power in large quantity, reckoned at the prime mover.

23 And in all these new schemes of superpower — St. Lawrence power or steam stations at the mouth of the mine — I see no chance for any substantial reduction in power cost at wholesale below that which we now enjoy in either the near or the distant future.

[1] This refers to average practice of the times — not to the few best plants. An admirable review of the state of the art of steam-engine practice was prepared by John C. Hoadley in 1884 for the meeting of the British Association for the Advancement of Science in Montreal, and printed for private circulation under the title of Steam Engine Practice in the United States, which gives several examples of engine economy close to present-day standards of best reciprocating engines. The great change is to the steam turbine in the central station, with single units 50 or 100 times as powerful as the common factory engine of 40 years ago.

Future Holds Small Hope of Cheaper Power

24 In a broad review and for a round figure we may regard the general station cost today, hydro or steam, as 1 cent per kw-hr.

25 One cent per kilowatt-hour is equivalent to $22.36 per horsepower year for 10-hour power drawn on 300 working days. Although I have no recent or broadly collected figures, and freight rates on fuel cause wide differences in cost, it appears probable that few textile factories in New England with the best large reciprocating engines are now obtaining their power for less than, say, $30 per hp-year, which would be about equivalent to 1.5 cents per kw-hr. for 9 working hours where all costs are reckoned in. I am told that the best large stations today can cut this figure in half, for cost at bus bar. It is estimated that with units of 30,000 kva. — steam at 600 lb., coal at about $5 — the cost at bus bar can be brought to 6 mills per kw-hr. under a favorable load factor, and there are a few great hydro stations at localities that are few and far between where it is reported power can be put on the bus bar at 3 mills per kw-hr., including all proper charges.

26 But on top of the station cost there must be reckoned a substantial cost of distribution, and there are large stand-by charges in the yearly load factor of a 9-hour day.

27 Let us not be too optimistic about cheaper power coming from St. Lawrence, or from coal mines, over the superpower line to central New England or Greater New York. I, for one, strongly incline to the belief that factories in Lawrence will be glad to get water rights at the old figure of $20 per hp. 50 years hence, and that 1 cent per kw-hr. (perhaps 1.25 cents) at wholesale in big blocks will be the lowest price or cost attainable by the great factories at Lowell and other Mid-New England industrial centers 50 years hence and in all the years between. The great benefits of the superpower system and of the distant sources will come in lessening the investment of capital by the industrial pioneer, and in many other ways than by reducing present costs per horsepower-hour.

The Start of a New Era

28 In general, 30 to 40 years ago water-power developments were moving slowly, waiting for the great impetus soon to be given, first by the enormous increase in the demand for cheap paper from wood pulp, and next by long-distance electrical transmission, followed, 10 years later, by demands of the electrochemical industries, particularly that of making aluminum from purified bauxite.

29 The U.S. Census for 1880, under the broad vision of its director, Gen. Francis A. Walker, later president of Massachusetts

Institute of Technology, published two big, thick volumes on water-power development, based largely upon field inspections by George F. Swain, Dwight Porter, Herman Hollerith, and others among our members since become famous, which was a thoroughly admirable review of the state of the art and extent of development in the U.S. 43 years ago, and was so instructive a piece of history that we may well urge that the next census prepare a similar volume which will set forth, with authority, the wonderful story of a half-century's advance in the manufacture of power, both water and steam.

30 Thirty years ago the greatest modern power consumers — electric railways and municipal lighting systems — had scarcely been brought into existence by the use of alternating current and long-distance transmission, and the great water-power sites in northern Maine, northern New Hampshire, New York, and Vermont had not been called largely into use by the modern demand for newspaper pulp. Thirty years ago the water-power company at Lawrence still had water rights for factories unsold, while customers were slow in coming. Along each of the chief New England rivers were undeveloped power sites, at least one of which still runs to waste.

31 I saw something of the pioneering work in high-speed engines, for I knew intimately John C. Hoadley, deep student, prophet of the great future of power, and most lovable of men, and was well acquainted with his shop and his assistants, Pardon Armington and Gardiner Sims, when they were building high-speed engines for the early small Edison generators. Sims showed me one of these running at about 1000 r.p.m. as the very end of things in speed. They put Hero's steam turbine on the title page of their catalog as typifying speed in steam power; far from imagining that its modern development in the great central-station turbine would spoil the business of building reciprocating steam engines and shut up most of the large engine works. I was a bystander at several of the early, tentative installations of the dynamos of Brush for arc lighting, and of Edison for incandescent lighting, as these were tried out in a hesitating way for lighting a few places in factories and streets.

32 How pitifully small those first plants now seem! And how wonderful were the developments of the next 25 years! The great Corliss steam engine, at whose gigantic frame we marveled at the Philadelphia Centennial — the dominating central feature of the great machinery hall — was only of about 1300 hp.! [1]

33 The ponderous steam engines used in emergencies, such as backwater or drought, at the Pacific Mills, the Atlantic Mills, and the old Washington Mills, at whose majestic proportions and giant power we youngsters marveled, were each of less horsepower than

[1] Steam pressure at throttle, 100 lb.; cylinders 40 in. diameter, 10 ft. stroke; 36 r.p.m.; cut-off at 25 per cent of stroke.

the Liberty motor, model of 1918, in an airplane, weighing 825 lb. and giving 450 hp.

34 Few of the largest water turbines in Lowell or Lawrence were of more than 200 hp. each. An 80-in. Swain wheel of about 800 hp. put in at Lawrence about 45 years ago was regarded as a monster, as "the most powerful turbine ever built," and finally its buckets (of cast-in plates) broke loose, unable to stand the strain of 30 ft. head.

35 The one turbine which I saw started at Pit River, California, last October, under about 450 ft. head, had more power (40,000 hp.) than all 65 of the Lawrence turbines combined, plus all of the Lowell turbines. And a single one of the recent turbines at the Queenston-Chippawa plant developed 60,000 hp., or more than the total obtained in all three of the old water-power cities — Lawrence, Lowell, and Manchester — from the Merrimac River, which we used to say was "the hardest-worked river in the world."

36 Things were moving slowly in water-power development and we felt we were near the summit of things, until two new factors came over the horizon: wood pulp and electrical transmission.

37 Steam felt the new stimulus 10 years before hydro woke up, but since that time a merry race has been on between the two, both in size of unit and in economy of service; first one winning and then the other, as circumstances favored, and the end is not yet.

38 The changes of 30 years are far less in the cost of power than in the scale on which it is developed, its widespread distribution, and its ever-increasing uses.

Power, Founder of a New Epoch

39 Twenty-eight years ago I heard George S. Morison, President of the Am. Soc. C. E., deliver his presidential address on the New Epoch in Civilization Opened by the Manufacture of Power. His statement was so strong and clear that its keynote has been ringing in my memory through all these years. Later this was combined with a Phi Beta Kappa address at Harvard and an address at Rensselaer Commencement, on other aspects of the same topic, and published in a little volume under the title, The New Epoch. Edward Everett Hale told me that Morison's was the most inspiring Phi Beta Kappa address to which he had ever listened.

40 President Morison developed his thesis that just as the several steps in the advance of human kind, from animal instinct, through savagery and barbarism, to civilization, had resulted successively from the use of fire, from the invention of the bow and arrow, the invention of pottery, the domestication of animals, and the manufacture of iron or bronze, until the epoch of civilization began with the invention of a written language, which stored up the observations and experience of one for the benefit of all who followed; so now a new and far-reaching epoch in civilization had

begun with the manufacture of power, increasing the power of the hand of man almost without limit, and making neighbors of all the world. And he argued that this new epoch had hardly begun, although the power expended by one of the great steamships of 25 years ago in a single voyage across the Atlantic was greater than that required to take from the Nile and place into final position every stone in the Great Pyramid!

41 Since Morison's time the advance has gone on with a stride that has far outstripped even his vision.

42 Kind and appreciative words should be spoken of the financiers of vision who have risked much and sometimes lost heavily in pioneering on these great machines by which the general welfare has been so greatly advanced. A high officer of one of the big concerns told me how more than $200,000 had been spent without success in developing a new steam turbine, and the whole seemed about to be sorrowfully charged off and the scheme abandoned, when after a long debate, just one more allotment of $10,000 was granted with the firm agreement all around the board that this should be the last — and this last-chance grant brought success.

43 The great hydraulic turbine has become one of the most marvelous machines ever fabricated by the hand of man, marvelous in the efficiency with which it can extract 94 per cent of the energy of a great falling current of water in the short space of 3 or 5 ft. of travel through its runner within less than one-tenth of a second of time! And it can do this 24 hours per day, 7 days in the week, year after year, with marvelously small cost for care and oversight. The wonder of all this came over me a few months ago in a hydroelectric plant of 30,000 kva. with *only two men* present to care for it all! Picture two men guiding and grooming not that famous 20-mule team, but one of 40,000!

44 Next day I got the superintendent at another plant near by to hoist the spillway gate so as to let a quantity about equal to the discharge of one turbine fall nearly a hundred feet; and watched its tremendous display of force in high foaming surges and tumbling eddies, in comparison with the quiet discharge of equal volume from which the nearby turbine had extracted the energy. Such a display of force appealing to the eye speaks louder than words of this triumph of engineering in controlling one of the great forces of nature for the benefit and convenience of man.

Change of Method in Solving Problems of Development

45 I recall with much interest that about 40 years ago, Sir Paget Higgs, of England, knighted for services in laying the first successful Atlantic cable and an interested student and prophet of electrical development who had become fascinated by the idea that when a direct electric current was led into an electrical generator it would run this backward and convert it into a motor, came to Lawrence to study its water power and the uses thereof in the

factories, chiefly to find out if there was important practical use for electrical transmission of power; and I had the good fortune to act as his guide along the power canal and through the Pacific Mills. In the evening I heard him hold forth at the chief literary club of the city to local captains of industry and business men. He was regarded as "interesting, but too visionary." The verdict was that there was nothing practical in his great idea!

46 The most fundamental changes in scheme for developing a great water power that have come in the past 30 years have come from the distribution of power by electricity. Then *water was distributed* to the turbines, now the *power is distributed* to the machines in the factory.

47 The scheme followed at Lowell, Lawrence, Manchester, Holyoke, and at most of our large old-time factories, of building a dam near the head of the rapids and distributing water by long canals, has been put badly out of date by electrical distribution.[1] Now canals are abandoned, the power house is built into the dam, notwithstanding the cost of the dam may have to be vastly increased by building it of greater height down near the foot of the rapids in order to use the whole fall; or if the dam is not placed at the foot of the fall, a deep, long tailrace is excavated instead of a canal.

48 Some of the most revolutionary changes are:

a Dispensing with a long power canal controlled by head gates by building the power house into one end of the dam and thereby lessening ice troubles and making efficient any surplus of flood head, instead of losing it by throttling in sluiceways and canal head gates.

b Higher heads on turbine — even to 825 ft. as against the common old 30- or 40-ft. limitations — are made practicable by stronger design.

c Dams are built to the extreme limit of height permitted by river bank and flowage rights, by means of larger tools and with greater confidence about the theory of design, so as to form the largest practicable storage reservoir and give both better storage for peak loads and flow, for Sunday storage, and for a drought reserve. Whereas, in the old days 30 ft. was about the ordinary limit in height for dam and 10 ft. the prudent limit for crest depth in floods, one now does not hesitate at water-power dams more than 100 ft. in height, and fears not whatever crest depth the flood may bring.

d The siphon spillway has become well established as a safe and efficient means for conserving the power of heads up to nearly the full limit of flowage height and meanwhile adding largely to the volume of storage available for regulation.

[1] How a new departure in the power layout of Manchester, N. H., was made two years ago, has been told in a professional paper by Arthur T. Safford, Mem. Am. Soc. C. E., published in Proceedings of the Boston Society of Civil Engineers.

e Turbines are made far larger than formerly, are given the highest practicable speed consistent with maximum efficiency, and the number of units at one site is made the fewest possible.

f The old-time ponderous bevel gears on top of the turbine, with their frequent breakage of teeth, are unknown in modern design; and the shift from vertical to horizontal axis of turbine for a more efficient attachment of main driving belts or generator, now is far in the background. Inside the factory great toothed gears, heavy lineshafts, and great belts for distributing power from water wheel or engine have mostly become things of the past.

g Nearly all friction loss of transmission from turbine to generator is avoided by placing the generator on the top end of a vertical turbine shaft, and notwithstanding enormous increase in weight of runner and rotor up to four hundred and eighty tons, for example, the troubles of the old-time foot step bearing have been done away with by top suspension, or by a bearing in which viscosity drags the lubricant under the load. Four hundred and eighty tons spinning around apparently lightly as a schoolboy's top, "asleep," surely is a marvelous triumph of engineering!

h Turbines are made taller and so designed as to obtain highest efficiency when at less than full gate opening, thereby providing a surplus of power, or overload capacity, for emergencies and speed regulation.

i The high-speed runner, trimmed down almost to the proportions of a ship's propeller, is establishing a firm place for itself in the economics of design for special conditions.

j The utmost care is taken in design to prepare smooth passages and easy curves for the water passing through, and a spiral approach for further lessening eddy loss is now universal. (It is worthy of note that Uriah Boyden, the hydraulic genius of 70 years ago, used spiral approach and top suspension in his best designs and made his price contain a bonus for each added per cent of efficiency obtained.)

k The so-called Leffel type of speed gate, regarded in the old days as too "trappy," too loose, and too leaky, has in these later years become glorified by accurate machinery into the standard type for all makers and up to the very largest sizes.

l The ideal of the automatic hydroelectric station for small powers is here in such perfection that after filling the oil cups on the bearings one can depart, lock the door, leave it in solitude for a day, perhaps for a week; starting, stopping, and regulating, all under control of the man who throws a switch or presses a button, miles away.

m The electric steam boiler has become an economic possibility for absorbing surplus power until profitable customers can be found. With steam coal at $10 per ton, the equation stands giving a value at about $13 per hp. per year for continuous 24-hour, 7-day power (the precise figure varying a little according to the respective efficiencies of the coal-heated boiler and the electrically heated boilers

compared). Correspondingly with \$5 coal, the equivalent power price would have to be about \$6.50 per 24-hr. hp. per year, or *less than the mere interest*, exclusive of maintenance, on an irreducible minimum cost of development. For heat processes, as, for example, wood-pulp digesters, the 24-hour demand is common and for such year-round, day-and-night work the electric boiler may be a great help in meeting interest charges during the early years. The hydro power could hardly enter the competition and earn dividends on a 27 per cent annual load factor, which corresponds to 8 hours' use daily for 300 working days per year, unless coal was costing more than \$40 per ton.

n The foregoing changes relate to the power-house design. Other changes of even greater importance to the builder of a new factory have been brought about by the distribution of power by electricity throughout all parts of the factory and throughout city and county. Now the central station and the public-service corporation relieve the founder of a new industry from raising the large sum necessary for developing a water power or for buying boilers and engines. His capital is thereby conserved for building a larger factory, or if the new enterprise is something of a venture, he can try it out with a smaller investment. Moreover the economies of large power units, of wholesale scale of operation, and the diversity factor in a widespread system permit him to buy his power cheaper than he could make it; and more than one large factory with a good steam power plant already installed has found economy in shutting it down and buying from the public-service circuits. (If he has a turbine he keeps it running merely to save fuel.)

o One of the most fundamental changes of all is that of the view of the public toward water-power development. Formerly its only interest was in welcoming the development of power because of the benefits from employment that followed in its train, and with little or no thought of public control. Now where any general public service is to be given there is oversight by public officials at almost every step in development, and it is entirely possible that an unwise control may work harm to the long-range public welfare.

49 This change has come through the development of widespread public-service power systems and their demands upon the natural resources of river flow, from the necessity of invoking of the state's right of eminent domain in order to create a great development, or sometimes by reason of title still retained by the public in the river bed and reservoir sites essential to power development.

50 A great awakening has been going on regarding the need of some public control, lest rights given up by the public, or possible to obtain only through legislative act, be capitalized at all the traffic will bear.

51 The principle has already been established in some states that "Values inherent in a public resource developed and used in

essential public service by an agency created by law for that purpose shall not be capitalized in excess of amounts actually expended in acquisition," and this seems right and for the best.

52 Good practicable methods for safeguarding the public from unreasonable rates and invested capital from cutthroat competition or political confiscation are being worked out through public-service commissions, which promise much better service and much lower costs than possible through public ownership and a management under political control.

53 There is not here time in which to go into detail about the reasons that have led to these changes, nor about the various dangers that lurk in unwise control.

Interconnection of Power Systems

54 In hydroelectric practice the facility of long-distance transportation of power is the outstanding achievement of the past 20 years, and one of the most interesting features is the way steam-power and water-power installations at widely separated places, each pumping current into opposite ends of the same line, have come to supplement each other, economizing both power and transmission, uniting and interchanging the variable power developed in the mountains or foothills from water which is plenty in springtime but scanty in midsummer, with steam power developed at the seacoast from seaborne coal or oil. The electric current flows with equal facility in either direction.

55 It will surprise many to learn, for example, that more power is sent forth from the Narragansett steam station in Providence from its 60,000-hp. steam turbine, back into the country over the widespread lines of the New England Power Company, than is brought to the industrial centers near the coast from the northern Connecticut River and its tributaries. Hydroelectric companies on variable streams less favored with reservoir sites are becoming connected up with steam reserves at the seaboard, and all are rapidly becoming interconnected.

56 Few persons not directly engaged in power transmission realize the extent to which this interconnection has been quietly going on. In Providence not long ago, within about ten minutes after a burn-out near the central station, power was flowing in from four neighboring cities, so that all went well. At Chattanooga last year I saw the great hydroelectric station completely shut down, and the whole flow of the Tennessee River held back half a Sunday for purposes of inspection, and meanwhile current was flowing in, sufficient for all demands, from the works of three or four other public-service corporations many miles away.

57 Just now there is some confusion of thought in the present popular interest in superpower schemes between "superpower" from great new stations and the *interconnection* of central stations, by which one station can help another and by which the steam

station — far more flexible than the hydro in taking overload — can help out the hydro station in low-water seasons, while the hydro, in its flush times, can save coal to the steam station.

Each Site Has Its Own Problems

58 Broad generalizations about water-power problems have more special exceptions than followings. Each prospective site for hydroelectric development has problems peculiar to itself, and on beginning an investigation the chief problem at any one site should be roughed out from the two different standpoints of

1 Profit to investors
2 Public welfare.

59 From each standpoint both a close-up view and a long-range view into the future are necessary; remembering that the present value of a power site rests solely on possibilites of its profitable use. There are many enticing prospects of water falls on flashy streams, which investigation will show to have only the value that pertains to beautiful scenery, for perhaps 50 or 100 years to come.

60 A site is of no more value for hydroelectric power than for a cow pasture except as its potentialities can be made to minister to industry or public welfare. And although its prospects may figure into many millions of horsepower-hours, one must first find the $100 or $200 per hp. to fit it to produce power, and before this one must know definitely what work there is that really needs this horsepower, what price it can pay per horsepower-hour or per kilowatt-hour, just how much of the power is "firm," or primary power, and for what part of the working hours, 365 days per year, this is surely dependable under adverse conditions of flood, drought, and ice.

Public-Welfare Problems

61 These go beyond the scope of utility commission in its supervision and limitation of scope of development, prevention of cutthroat competition, protection against excessive charges and safeguarding good service; and reach out into considerations of conservation and promotion of favorable environment for good citizenship, and are up to the statesman rather than to the engineer; but the engineer must be on the alert to aid the statesman with facts.

62 From the point of view of public welfare no sane man can question that it is better to utilize water power now running to waste than to be exhausting the future's store of coal and oil, and therefore it is plain that the Government, in doubtful cases, should turn the scale, if it can, by all reasonable helpfulness. To plead for this we must know how great are the differences in costs between water power and steam power that must be overcome.

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tries scattered throughout the country is better for community life than crowding everything into great cities. In this aspect of industrial development — country village vs. metropolis — electric transmission seems thus far to have been hurtful rather than helpful.

64 Let us look into this carefully and perhaps we shall come to sympathize with what both Maine and Canada are striving for in their restrictions upon the export of power, although neither may have yet found the best solution for its fundamental problem.

65 Surely, as good citizens jealously guarding the future of our country, we should study these hydroelectric problems from a longer view than that of merely capitalizing a power site for the quickest return to that promoter who wakes up first to its value.

66 Let us hope that some one with an extremely long range of vision can find justification for the millions of taxpayers' money recently expended at Muscle Shoals.

Value of a Water-Power Site

67 As to the value of an unused power site, or "mill privilege," as it was called a half-century ago, the public is mostly misinformed, or uninformed. Commonly there are two sides to be considered and weighed in all of those cases where public ownership of water right river bed or reservoir lands is needed for the development.

68 Such cases may raise four important problems:

1 What price should the state receive (*a*) from a public-service corporation under state supervision as to rates and service; and (*b*) from a private manufacturing corporation under no particular restraint? Should there be a difference?

2 Should the state block the enterprise until the time is ripe for a development by public funds, owned and managed by public officials (i.e., political control)?

3 To what extent should the state promote hydroelectric development by public or by private means (*a*) for the purpose of conserving a supply of coal and oil for posterity and (*b*) for the purpose of later developing new industrial communities in country districts, away from the great cities?

4 The conservation of the limited supply of capital and labor for use where it will do the most good.

69 Those who represent the public in legislative halls often seem possessed of hazy and ill-defined views on all of these matters, and engineers may well give liberally of their time in a campaign of education as a highly useful patriotic service.

70 There is a widespread notion that almost any waterfall can be turned by hydroelectric development into a never-ending, ever-flowing stream of gold, with small understanding of the amount of capital that must be buried beyond recall in such a

venture, and on which interest must be paid or income found if capital is to be conserved.

71 Those who talk of the public's being robbed of its birthright by turning over for a nominal price or rental the potential resources of flowing rivers at rapids or falls to great, soulless corporations, mostly seem to have done neither clear thinking nor accurate estimating, nor to have made a close study of any particular real prospect. There may be circumstances where if a new industrial center or community "in a green country town" can be founded by coaxing capital to take the risk of a new power development, the state or nation may wisely remit all rental price for whatever the public owns that stands in the way, from motives like those that lead enterprising boards of trade to offer free sites and even subsidies to attract industries to their town.

72 Nor do many of these who talk loudly about development by public funds from taxation and operation under public officials as a source of profit to the public or as a means of stimulating industry by cheaper power, have any conception of the rapid rolling up of compound interest upon those parts of a large investment for whose product there is no immediate customer.

73 In the great early American power developments at Holyoke, Mass., Lewiston, Me., and Cohoes, N. Y., the original investors are reported to have lost all or the greater part of what they put into these developments, because of fixed charges and operating costs accumulating while waiting for customers to absorb the output. I heard the venerable Charles S. Storrow, treasurer and first engineer of the great water-power development at Lawrence, Mass., testify that they "paid one dividend on expectations and then passed thirteen on facts," while waiting to sell that portion of the power left over after supplying the three great factories built simultaneously with dam and canal.

74 About 15 years ago, after I had made a careful study and report on the possibilities of power development on a great American river, I was called before a group of about fifteen prominent financiers who had met in the Wall Street district to consider the promotion. My cross-examination was begun by substantially this statement: "Mr. F., there are today 30 to 40 million dollars of questionable water-power bonds floating around this financial district on which it is difficult or impossible to earn the interest; such, for example, as the X Company, the Y Company, and Z Company, etc., etc. (giving names then prominent). For each of these there was a rosy prospect at the time of the underwriting. Now, will you kindly begin by explaining wherein this present prospect differs?"

75 The answer was to be found in the delays in selling, not in mistakes of engineering. The trouble came in the period that those several projects had to wait before selling the last half of their power.

76 Hydroelectric development differs from steam-electric development or other power development in that *the big expenditure*

for water power comes mainly all in one bunch, whereas with steam it comes step by step, as needed, for almost immediate use.

77 In water-power development the site, the water rights, the flowage, the dam, the power-house foundations, and the larger part of expenditure for turbines, electric apparatus, switchboard, and transmission line must be made in full at the start, and if customers are not already signed up for sufficient kilowatts to pay fixed charges, the prospects are uninviting.

78 The investor will do well to look backward while the promoter is trying to lead him forward, and study some of the financial wreckage.

Steadfast Flow and Ready Demand

79 Lack of steadfastness of the water supply spoils many hydroelectric prospects. Ofttimes the investors in a water-power development have been pained to discover later that they must add a steam plant to their investment. Commonly a prospective dam site on a flashy stream has no substantial cash value, save where it can be developed by electric transmission as a mere "coal saver" to some widespread utility system that already has a full quota of engine power, and in such cases the margin of profit on development commonly is not large. The Great Falls of the Potomac, within 15 miles of the dome of the capital, still remain undeveloped, though often investigated; but doubtless their day of use will come as a coal saver to an existing steam power plant in the nearby city. The McCall's Ferry development, now called Holtwood, on the great Susquehanna, was a sorry financial venture until tied up to the steam stations of Baltimore Public Service.

80 On a variable stream it may be that great reservoirs can be built to regulate the flow, but there the large cost of the reservoirs has to be subtracted in determining the value of the power site. Or, it may be where river banks are favorable and not preoccupied by railroads, that, with the increased engineering skill and courage of recent years, a very high dam can be built at a cost of a few million dollars, to create a reservoir close to hand.

81 The point which I would stress is that, when one looks at a waterfall or rapid, "all is not gold that glitters."

82 There are hydroelectric sites with many millions of kilowatts of capacity, in the aggregate, still awaiting development, from Maine to California, from Canada to Mexico, but to decide which ones are worth while, and which ones will be merely sink holes for capital, is a painstaking task.

83 There are today still many magnificent opportunities for power development lying around unused that are practically without commercial value simply because there is today no use or prospect of early sale for their possible output, and there are other vacant opportunities for hydroelectric developments, figuring into millions of kilowatt-hours per year, which have no large present

commercial value because of the large cost of supplementing the low-water flow by steam, or because of the vast cost of establishing a reservoir by high dam or otherwise for regulating the flow.

Demand by Chemical Industry

84 There is a widespread popular misunderstanding about the present demand for water power. Many think there are great chemical industries earnestly seeking hydroelectric power. The chemical engineer of large vision is today looking for cheap B.t.u. far more than for cheap kilowatt-hours.

85 This statement may seem strange to one familiar with the vast electrochemical industries at Niagara or with the vast use of power for making aluminum out of purified bauxite, until he has done considerable investigating or has tried to find a customer for a big block of power in the chemical industries. Noting that nearly all those great electrochemical industries making aluminum, caustic, sodium, chlorine, cyanide, carborundum, etc., which are now found located along the Niagara Gorge, rest on processes discovered since capital first had courage to develop Niagara power in advance of customers in sight, one might be inclined to prophesy that another similar great development would similarly be absorbed by electrochemical industries.

86 Before thus prophesying he had best ask just what new large demands of similar kind actually have come during the past 10 or 15 years from the hundreds of laboratories all over the world in which able chemists are making research.

87 A prominent and successful Canadian chemical engineer told me that at the close of the World War he sought to establish himself in some large line of manufacture of chemical products, and familiar with the vast undeveloped water-power resources in Canada, upon rivers other than the St. Lawrence and requiring smaller first outlay, he sought far and wide for chemical uses for this power, but each time the trail led him away from the kilowatt-hour to the cheap B.t.u. In other words, to cheap coal, and finally he located in the coal fields of West Virginia.

88 I know of several owners of power sites of great potentiality who have employed able chemical engineers to seek far and wide in science and industry for uses to which these prospective developments might be put, but still each of these rivers, as Bryant said of the Oregon, rolls on, "and hears no sound save his own dashings."

89 I know where, in Canada, a million horsepower can be developed in blocks of not less than 100,000 hp. at almost unprecedentedly low cost, perhaps $100 per kw. of primary power, if only profitable use could be found for blocks of this size; but until the chemical engineer comes forward with new uses, as perhaps in iron smelting, or new economic processes for fertilizer, or until steam coal becomes worth $15 per ton at great business centers, I fear such sites have no present substantial commercial value.

90 And as to the demand for big blocks of cheap hydroelectric power to be used in the manufacture of cheap fertilizer, the public and their deputies in legislative halls have become badly befogged and misled, particularly in the discussions about the development of cheap water power at Muscle Shoals. And a fuller explanation may sometime be in order from the U.S. Army engineers or whoever is responsible for putting in about double the turbine capacity that ordinary business considerations would call for on a river of this small low-water flow, or that business concerns have put in proportional to flow, elsewhere on the Tennessee River and on neighboring streams. This great Muscle Shoals development seems to some of us to be mainly the result of lack of information and diligent propaganda, of a desire to "help the South" — sending good money after bad while hoping to find cheap fertilizer at the end of a rainbow.

91 The public is mostly unaware that new inventions have lessened the relative importance of cheap electricity for manufacturing fertilizers. Our ordnance officers knew this long before the Muscle Shoals hydroelectric development was begun and adopted a modification of the German "Haber Process."

92 The old spectacular processes for the fixation of nitrogen developed by German science and capital in Norway a dozen years ago, with flaming electric arcs 10 ft. long and of 2 per cent efficiency, were long since outclassed by processes in which a mysterious catalyzer, under favoring conditions of great pressure and high temperature, is doing the work with a relatively small call for power. The nitrogen research bureau of the Federal Government with the vast sums they have spent on research should be able soon to clear the foggy atmosphere that now surrounds the relation of cheap fertilizer to water power.

Value Depends on Use

93 I have made the previous statements to strongly fix attention on the fact that the value of a prospective hydroelectric site is *not in the water and its fall*, but *solely in the use to which this power can be put* and the rate at which the total output can be absorbed in industry, so as to pay interest upon the capital invested.

94 If, in a great opportunity for hydroelectric developments, one can obtain firm contracts for, say, half the "firm" or primary output at such prices that he can be assured of bond interest and operating expenses, perhaps he can find capital sufficiently optimistic in faith in the future to wait for its dividends on that part of investment represented by stock. The investor's hope of "velvet" in general must rest mainly on expectations of future sales and on growth of the country, growth in habits of luxury, more illumination, and performing more and more of menial tasks by electricity. All these surely are coming. One who studies the yearly load curves of any great system will marvel at the rate of increase in

current consumption from year to year in a district long since seemingly equipped to saturation. But the engineer must seek through most careful analysis to learn *just* where and *just* how fast the increase on his "prospect" is coming.

95 On the other hand, many who urge conserving nature's fuel resources for posterity seem to have not weighed the importance of also conserving capital for more immediate needs, and the importance of investing the limited supply of available capital where it will do the country the most good.

Conservation of Capital

96 This frequently is forgotten by those who most loudly urge conservation of natural resources.

97 Conservation of capital always should be considered along with the conservation of natural resources throughout the design. This frequently will lead to a departure from the line of engineering development that would be ideal at a given site, because of the menace of an unsatisfied interest charge upon capacity provided but not promptly put to use.

98 It is painful to a good engineer to see only the most cheaply developed part of a great power site cut out and occupied. Sacrificing part of the fall, or sacrificing much of the flow of a variable stream for eight or nine months of the year, may be repulsive to one's ideals, yet frequently there is no other way in which financial loss to the investor can be avoided. Stating the case in another way, unless this is done, development at any profit must be definitely postponed, and all of this water and fall continue to run to waste. In such cases the *rights of posterity to the full conservation of this natural resource should be guarded by a time limit in the franchise*, with provision for review at the end of the term.

99 A sinking fund will sometimes afford a remedy and works may be laid out on such lines that after 20 or perhaps after 40 years the saving made at sacrifice of full utilization, in view of present high rates for procuring capital (often amounting to 8 per cent or even 10 per cent after brokerage and interest during construction), will, by compounding to the end of this period, permit of highest conservation. The interests both of the country at large and of the investor may be manifested by such a course.

100 Having made the foregoing statements regarding site value, public welfare, and conservation of capital by way of introduction, we may now consider in more detail a few of the fundamental problems in hydroelectric design and the questions which these involve. Nearly all of these questions lead to, or center around, economics.

Some Preliminary Problems

101 When a site is presented for consideration, the line of analysis ordinarily is as follows:

1 What is the maximum power in kilowatts that the site can be made to produce economically, and the number of kilowatt-hours per year?
2 What is the size of the successive steps, in which this total development may best be made?
3 What is the accessibility of the site; its distance from established centers of industry, or of population; or its proximity to raw materials that can be manufactured by its use?
4 How variable is the natural flow of water and how variable the fall, day by day, and year by year; to what extent can this be regulated?
5 What are the requirements of stream reserve, or the needs of interconnection by long-distance electric transmission lines to other sources, in order to provide for given amounts of primary power, i.e., power that can be depended upon day by day, year after year?
6 What are the prospective load factors, daily, weekly, and yearly?
7 At what rate will the present or prospective market absorb this output? How large a paying load can be taken on at the start? How many years will be required to absorb the entire output of the first stage of development? How long will it take to absorb the entire output proposed?
8 What particular form or sequence of development will be most profitable, sacrificing (if need be) efficiency to lessen first cost, but with works so laid out that the savings thus made, if put into a sinking fund, could ultimately be used for complete development of highest efficiency?

Centralizing vs. Scattering Industries

102 Hydroelectrical development of the past 25 years presents a fascinating picture, but let us not forget there is something in the nature of a seamy side. Its work so far has mostly been in methods opposite to those by which in the old days a water-power site became the center of a new community under conditions that make for wholesome home life and good citizenship.

103 The country life and the citizenship on which our future must be established are found at their best in those small cities and villages of the New England, Middle, and Mid-West States, built up around industries driven by the small water powers. Brunswick, Maine; Exeter and Claremont, New Hampshire; Springfield, Vermont; Fitchburg and Orange, Massachusetts, Thomaston and Waterbury, Connecticut; Dayton, Ohio — one could name by the dozen communities founded on water power while going over a map

of the older states and out to the "Western Reserve," even to Minneapolis and Lawrence, Kansas, to Spokane and Oregon City.

104 Although, as stated a moment ago, water-power development got a tremendous impulse about twenty-five years ago from an electrical distribution of power and the possibilities of carrying the power over a long distance for immediate sale in enormous blocks to replace steam power in public service of lighting, street railways, and power for minor industries, thereby assuring immediate income sufficient to carry fixed charges, hydroelectric power has built up a new industry in but relatively few places, and seldom has it been the foundation of a new neighborly community. Seldom has a hydroelectric development found employment for more men; so far it commonly has lessened the total number of men employed, when miners and stokers and ash handlers are counted.

105 While this reaching out from a prospective water-power site to the established steam central station of some distant city is surely the best road to a quick income, we may well pause and ask if there are not many cases where the future of our country would be better served by keeping the hydroelectric power at home and providing it only with short transmission lines for driving local industries. Sometimes it may be best for public welfare in the long view to make haste slowly and wait for the industry to appear or for the local "Board of Trade" to find it.

106 Have we not been led too fast by promoters desirous of a quick profit or by manufacturers who did not see the preservation of old-time American standards of greater importance than the temporary success based on cheap immigrant labor?

107 Electrical transmission of both steam and hydro power is now helping greatly in the economical establishment of new industries, but up to date it has been favoring further concentration in cities. It would be well if something could be done to start a movement in the opposite direction. As the network of power lines becomes more and more interconnected with the smaller communities, giving to them an ample supply of cheap and dependable power, many kinds of industry might again turn to the country village, attracted by lessened real-estate values, fewer strikes, lesser taxes, and by those benefits difficult of precise appraisal that come from the wholesome conditions of family life and social privilege that may be found by operatives in the smaller communities. A wire can carry current in either direction, and there are many cases where favors could be shown by the public in smaller burdens of taxation, etc., etc., that might turn the scale. While things are moving rapidly in relation of the public to power development, harm may come if those who lead the movement are not well informed.

Education in Public Relations

108 As the speaker said in beginning, these problems of public relations are becoming the most fundamental of all the problems of hydroelectric development.

109 After prolonged taking of testimony and discussion we now have the Federal Power Act, permitting and encouraging power development from rivers flowing through the public lands of the Great West. Before obtaining this some very intense misunderstandings had to be cleared up. It seems to be working well.

110 The Ontario Power Commission is trying out public ownership and distribution of power on a gigantic scale, and its results have been widely discussed, pro and con. Perhaps it is too early to judge of it wisely until its secondary effects in building up industries are better established.

111 The San Francisco City Government recently has gone on record 13 to 2 in favor of municipal distribution of 40,000 hp. of Hetch Hetchy power, with but small understanding of what this would cost, of the influence of varying loads, or the requirements of steam reserve in the city, or the needs of interconnection for emergency service, or of how all these could be accomplished for the best interests of all concerned. The speaker was challenged when there recently by a reporter out for "a story," to defend, if he could, the outrageous greed of a corporation that would offer the city only a cent per kilowatt-hour at wholesale, while he (the reporter) was charged 8 cents at his home. He could not believe that their present power companies in the San Francisco district now or recently had sold power to each other wholesale at three-quarters of a cent per kw-hr.

112 One of the great problems of water-power development is the education of the general public to understand the costs added by overhead, by steam reserve, by peak loads, by stand-by charges, and by distribution to the small consumer, by the expenses of measuring his draft, collecting his monthly account, and of developing in large blocks safely ahead of the demand, by maintaining a readiness to serve during drought, storm, flood, or fire, by elaborate interconnections with various widely scattered plants.

113 California, at about the time of last year's elections, was shaken more than by any earthquake over the question of whether or no the state should bond itself for 500 millions of dollars for development or acquisition and distribution of hydroelectric power, and although this was defeated 2 to 1, those whose seismographs are in adjustment say the same question is soon coming up again.

114 Some years ago the governor of a great state, himself a man of small vision, blocked the beginning of a new industrial city in a sparsely settled corner of his state by canceling the terms of a lease of river bed and dam site to a great power company, on the grounds that the price per horsepower agreed upon with preceding governor of the opposite political party was insufficient, and that

the public was thereby to be robbed. He seems to have failed utterly to have vision of the benefits that come from changing a cow pasture, remote from town or village, into a hive of industry, surrounded by a thousand cheerful homes, and of the risks to be run by some millions of capital in making the proposed development. Thereupon the corporation established its industries in another state.

115 New York State has by recent referendum voted 2 to 1 against permitting a microscopic proportion of its state lands to be flooded for water-power reservoirs and by this act also has forbidden place and passage for power house and transmission lines upon its forest preserves; an action strangely at variance with the recent policy of the National Government. There appears to have been an inseparable mixture of motives in this large adverse vote. The nature lovers seem to have been inflamed by wealthy owners of private forest preserves — and with some show of truth — against permitting such frightful blots on a beautiful landscape as have resulted in Maine and elsewhere from flooding forest or sprout land without first cutting and burning trees and brush to within a foot of the ground; but the most powerful motive seems to have been a broadly cultivated idea that this defeat would preserve a great opportunity for water-power development under state ownership, with *no real understanding of the facts by the voters at large.*

116 Ten years ago I worked with the New York State Water Supply Commission diligently for two years in organizing surveys for the fostering of reservoir development by the state for aiding the development of power, and from intimate knowledge of some of these vast opportunities for saving fuel and contributing to industrial development, it is unbelievable that the state will keep them in idleness many years.

117 The delay will not be all to the bad if it is employed to work out plans for making these power sites contribute to building up small, scattered communities. Using the reservoir water solely for pulp grinding at the rate of 100 hp. more or less per workman is not its highest use, neither is the mere transmitting of it to save coal, or to supply factory power, in cities already far too big for good living conditions.

118 Recently our Federal Government has shown a most wholesome interest in these large problems of hydroelectric development by promoting the Eastern "superpower survey," and recently Secretary Hoover, of the Department of Commerce, has given this important matter a new impetus. Those problems will be dealt with by the speakers who follow, and I hope the distinction between a superpower system and interconnection may be made clear, but I cannot refrain from quoting briefly from Secretary Hoover's inspiring address before the public utility commissioners of ten eastern states, in New York a month ago. He is reported to have said:

Engineering science has brought us to the threshold of a new era in the development of electric power. . . . We can now undertake the development from cheaper sources farther afield. . . . We can effect great economies through the interconnection of local systems. . . . We can assure more security from the effect of coal strikes and from interruption of railroad transportation. . . . Engineers report that more than 40 per cent of the railroad mileage of these eastern territories could be electrified at substantial economies of operation . . . the indirect results both human and material are even more important . . . One of the first principles is the free flow of power across state lines. . . . Physical barriers and other conditions naturally divide the United States into several distinct zones, one embracing New England and the Middle States, another west of the Alleghanies, etc., etc. . . . The problem of each of these sections of the country is different and must be solved separately.

119 At the recent Richmond Convention of the American Society of Civil Engineers there was presented an extremely interesting and instructive talk by Mr. Wm. S. Lee, Mem. Am. Soc. C. E., on the present widespread interconnection of power systems in the Carolinas and other southeastern states, showing how this widespread benefit of a superpower system came mainly from the transfer of power in a great chain of systems *not by a long leap from one end of the system to the far-distant end,* but by passing surplus from one system merely *to the next in line;* in time of flood, drought, or breakdown, or in finding temporary use for the surplus of a new plant in helping an overloaded neighbor until he too could build larger, and how all of this was in the interest of the public.

120 *Can any reasonable man, familiar with the facts and with human nature, doubt that these far-flung developments can be most rapidly advanced and best carried on by great corporations, under a carefully guarded governmental supervision,* better than as an enterprise wholly in state or federal hands and largely dominated by politicians?

121 Things have now come to the stage where the most fundamental problem of hydroelectric development is the education of the public by making plain the facts.

What Next?

122 Today it seems that we must be close to the summit, in size of power units and in economy of performance, in both hydroelectric and steam-electric; with line voltage of 220,000; steam at 1200 lb. pressure; water wheels under 3000 ft. head; hydro turbines at 840 ft. head, and single units of 60,000 hp. in water and 80,000 in steam. Surely 94 per cent efficiency on a hydro turbine can never be excelled.

123 At the Chicago Meeting of the Am. Soc. C. E. last July, Mr. Abbott, of the Chicago Edison Co., who perhaps has supervised the sending out of more kilowatt-hours than any other one man in the world, figured out that it was cheaper to haul coal in cars than to take power over a wire from a station at the mine where the distance was more than 100 miles, at 75 per cent load factor

and Mid-West coal costs; and presented a table showing that as between steam plant in city, with coal coming 200 miles on cars and a total of 100,000 kw. capacity costing $12,000,000, and a hydro plant with 200 miles' transmission costing in all $25,000,000, the cost of power delivered, in fuel and overhead for the steam plant, would be about 3.5 per cent smaller for the steam plant than for the hydro — under Mid-West coal prices and conditions. Mr. Junkersfeld, speaking after Mr. Abbott, held out prospects of cheaper long-distance transmission at 300,000 or 350,000 volts within 10 years, and Mr. Hamilton, also of great experience, said in effect that very few hydro developments have been made that can compete with the 5-mill power cost in the best and most favorably placed steam stations.

124 The only pathways open for further advance seem to be in higher pressures for electrical transmission over distances more than 250 miles, as from the St. Lawrence to New York City, and we have heard rumors that direct-current transmission has some things in store for the future, and then there is the possibility of some new and more efficient form of heat engine.

125 Who can tell how long the call for this long-distance transmission is to be held back by the success of the mercury-vapor turbine and by the success of those new steam turbines at 1200 lb.? I think it was old Hosea Bigelow who said, "It's dangerous to prophesy unless you know." Just what do we now really know?

126 The fact that our best reciprocating engines take out or deliver only about 16 per cent of the energy in the fuel, the best of the recent giant steam turbines only 18 per cent to 19 per cent, and the internal-combustion engine perhaps at best less than 40 per cent, offers plenty of scope for fame and fortune and for that joy of conquest which is irresistible. We are told that theory and tests of the first fairly large machine show that the mercury turbine in combination with its steam-turbine partner will cut coal consumption per kilowatt-hour nearly in half.

127 We have seen the "triple thermic motor," the gas turbine, the Humphrey gas pump and others start forth with rosy promise, stagger and fall by the wayside; but the present case is far from hopeless and the future full of promise in two particular directions.

128 Just now the mercury-vapor turbine in combination with steam turbine of 3000 hp. at Hartford [1] divides attention with steam put into a turbine at 1200 lb. per sq. in., or as near to red-hot steam as containers of known metals can withstand.

129 Will the next great advance come from these improved heat engines or from improvement in long-distance electrical transmission? This brings us forthwith into the discussion of transmitting power to Boston and New York from the St. Lawrence.

[1] As showing the advance in scale of experimenting, it is worth noting that the horsepower of this mercury-vapor turbine at Hartford is about double that of the great Corliss engine at the Centennial Exposition of 1876. For description of this turbine, see *Electrical World*, Oct. 13, 1923.

St. Lawrence Power

130 As to bringing St. Lawrence power to Boston, one of the most experienced among the builders and managers of large electric-power developments in America, an engineer who is familiar with electrical power developed in the large way from both water and steam, has recently told me that as a result of careful investigation he believes there is no money to be saved by bringing hydroelectric power to Boston from the St. Lawrence. He finds that turbo units of 30,000 kva. using steam at the recently proposed pressures of from 500 to 600 lb. per sq. in. at location on the Massachusetts seacoast, with coal even at the high prices of 1923, can deliver power to the bus bar just as cheaply as power can be delivered into the Boston circuit from a water-power station on the St. Lawrence, and with the advantage of less risk to interruption, and with the further advantage of smaller initial capital investment; avoiding particularly that in a transmission line for 200,000 kw., which, with its right of way, would itself cost about $50,000 per mile, or from ten to fifteen million dollars at the start!

131 Other very eminent engineers and financiers are just as strong and clear in their opposite belief, that under present-day conditions, *when account is taken of possible sales along the way*, in a block of, say, 200,000 kw. (perhaps smaller), power could be delivered to the Boston district from Canadian water power at substantially less than the cost of steam-electric power with the most modern type of plant.

132 Two separate groups of financiers and engineers for some months past have been independently studying ways and means and details for developing and delivering Canadian power, one group proposing to bring power to greater New York district from near the Long Sault on the St. Lawrence, while the other group proposes bringing power to the Boston district from the Ottawa River, close to its confluence with the St. Lawrence. And it is stated by a representative of one of these groups that they can put their hands on the necessary funds whenever the international and interstate questions of law and franchise have been satisfactorily settled.

133 The bringing of power into New England from even farther back in Canada has been actively discussed. It has even been proposed to transmit the surplus available from the Saguenay development at Grand Discharge, after satisfying the local demand for about 200,000 hp., in the vast pulp and paper mills now being built there, for both mechanical power and the generation of steam for all requirements, by water power in electric boilers.

134 Obviously, with so vast a local use ready to meet the fixed changes and maintenance of a power that can be developed in large units under high fall at abnormally low cost per horsepower, and with a large surplus flow of water, said to be enough for 200,000 hp. beyond local requirements, these rare circumstances may

present a condition of a great block of power so abnormally cheap that an abnormally large percentage of loss in transmission may be allowable, and that an abnormally high transmission from water otherwise wasted over the spillway.

Propaganda and Fog

135 Within the past two years there has arisen a great propaganda for power development resting upon factors far outside a strict comparison of solely the cost of water power in blocks of 100,000 or 200,000 hp. carried nearly 300 miles, compared with cost of steam-electric power developed nearby at the seaboard.

136 The farmers of the Western United States and those of Western Canada have united in a plea, which politicians, Congress, and Parliament perhaps may be unable to resist, for a damming and deepening of the St. Lawrence for the double purpose of the passage of ocean-going wheat ships from Lake Superior to the Atlantic, and the development of vast quantities of water power.

137 It seems that the farmers expect the water power to so largely reimburse the governments that the improved water transportation would be mostly a clear gain to the two nations sharing the cost of dams, canals, and dredging. Of course they assume that like the Canadian canals and the American canal at the Sault Ste. Marie, there would be no tolls.

138 Since the price of wheat from all over the world practically is fixed in Liverpool, the farmers hope that with the cheapened transportation thereby given, and with no tolls, this increased facility and reduced cost of shipment would give them a 10 per cent increase in the selling price of their wheat in the home town.

139 Meanwhile, also, the power promoters appear to hope that, with the two governments once firmly committed to financing (or guaranteeing) this vast project and with the money thus placed beyond recall, the opportunity for development of power created incidentally at the new dams and canals around the rapids, would after a time be leased at an almost nominal charge, rather than allow these great national resources to run to waste; and that thereby the vast surplus beyond local needs of water power from so cheap a source could easily bear the burden of 300-mile transmission.

140 Obviously, if the taxpayers at large of the two countries are willing to foot all the bills, and omit all ship tolls *in addition to increasing the price of bread for their own people* in proportion as the farmer gets nearer the Liverpool price for wheat, this would be fine for cheap local power and at small risk to the farmer. And *perhaps* ways can be found to bring this so cheaply to greater Boston and greater New York that it will cost delivered, say, one-fifth ($\frac{1}{5}$) of a cent per kw-hr. less than steam power generated at Fall River, Boston, or New York from sea-borne coal or oil. This exceedingly small fraction — about one-eighth ($\frac{1}{8}$) part of the factory cost of power, or about one fiftieth ($\frac{1}{50}$) part of what the householder pays for his

lamp light, seems about all there is in it for the average man in Boston or New York. The public mostly fails to realize that the greater part of the cost of electric power becomes attached to it after it leaves the bus bar or the central station.

141 There has been a lot of designing in outline, and much estimating and reporting to certain large corporations, and it would be extremely interesting if the details of these plans and estimates were open to public examination and intelligent criticism. The great captains of business, banking, industry, and transatlantic shipping who dominate the chambers of commerce of Montreal and New York seem far from being convinced that the published estimates of cost of all of this extensive river improvement are on a sound basis, or that any such benefit to the farmer in increased price of wheat could come, although no tolls whatever were imposed for the use of these enlarged canals. Indeed, some say that a saving has been promised greater than the entire present cost of wheat shipment from lake to seaboard.

142 So far as I can gather from personal discussion with business men and engineers familiar with the situation, the general impression is that last year's report of the International Commission was far from being so thorough a study as should be made before either nation accepts the proposed plans or estimates as final. Some believe that the actual cost would be double that estimated. Surveys and borings and other new data on cost were meager, and from personal observation I am confident that the alignment of this proposed new waterway was in some localities far from being the best for public interests that a thorough study would disclose.

143 Whatever may be the present obstacles to developing St. Lawrence power, I believe it certain as the sunrise that sooner or later they will be overcome; and it is plain beyond all argument that it is better for the two nations that this power be used to conserve the coal supply for posterity instead of running to waste. Nevertheless I fear that 10 or 20 years must pass before any important economy in power cost in New England or around New York City can come from this direction. High-pressure steam, cheaper coal, and perhaps the mercury turbine and the Diesel engine, may delay the day for service 300 miles away.

144 A great captain of industry told me a year ago that cheap hydroelectric power from the St. Lawrence is ultimately a factor of highest importance in the retention of New England's prosperity as a great workshop for home markets and export trade, and likewise for greater New York. And when electrochemists and physicists have performed what we expect of them, Montreal and Quebec may become the world's great center of electrochemical products exported to all parts of the world!

145 There is more than five million horsepower now running to waste along the St. Lawrence. It is hard even for an engineer to comprehend what five millions of horsepower of energy means, and difficult to translate it into the terms of every-day life. This is

about the sum total of electrical generating capacity today in all the central stations, steam and hydro, of New England, New York, New Jersey, and Pennsylvania.[1] And this St. Lawrence power being 24-hour power with ample storage for conservation, is equivalent to providing about three times the number of kilowatt-hours per year now used in lighting, railways, manufacturing, chemistry and metallurgy, etc., throughout this vast industrial and commercial region.

146 How soon do New York and New England need it? What will it cost here? How much can we pay for it? How much of it is needed here? These are its fundamental problems here.

147 In the neighborhood of Montreal, Cornwall, and Ogdensburg, with short transmission and more costly fuel, the problems are of different order.

DISCUSSION

[Formal discussions of Mr. Freeman's paper were presented at the session by Col. J. P. Hogan, representing the A.S.C.E., Geo. A. Orrok, representing the A.S.M.E., and Harold W. Buck, representing the A.I.E.E. Their discussions and an abstract of the discussions submitted from the floor follow.]

THE STUDY OF HYDROELECTRIC POSSIBILITIES

By John P. Hogan[2]

Mr. Freeman has very ably presented to you the fundamental problems of hydroelectric development and I have been asked to discuss that phase of the problem which is concerned with the need of accurate engineering data before proceeding with the development. It would seem almost unnecessary to emphasize the need for careful and accurate investigation if it were not for the fact that there are in this country today a large number of hydroelectric developments which have been financial failures. In fact, the number is so great that it is with considerable difficulty that independent hydroelectric projects can be financed. There are in addition numerous hydroelectric projects and extensions to existing systems which have failed to realize expectations and which have been carried through to a conclusion only by the existing financial strength of the companies undertaking them, and these, too, must be classed as failures from a financial point of view. Some of the failures in independent construction have no doubt been due to disappointment in the market but, in general, the majority

[1] According to the diagram and table published as a supplement to the *Electrical World* of Nov. 17, 1923.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.C.E.

of the failures in this class and almost all the failures in the case of expansions to existing systems have been due to two causes:

1 Overestimation of the available water supply, and

2 Underestimation of the cost.

Within the past year I have heard several executives of existing companies state that some of their water-power plants have failed to come up to expectation in power output and I have heard others complain that they had been misled, in some cases grievously, by the cost estimates of their engineers. I have heard a member of a banking house engaged in financing hydroelectric projects state in an offhand way, "I always double the estimates of the engineer." These remarks relate only to projects that have been pushed to completion.

In the experience of the consulting engineer there are for every project which has been completed twenty or more projected and receiving more or less consideration in the engineering and financial world, and in many of them the same faults of overestimation of water supply and underestimation of cost appear. In a promotion the reviewing engineer expects that a rather rosy view of the prospects will be taken by the promoter — and even in numerous cases by his engineer; but in the case of extension to existing systems it is more difficult to understand why such mistakes should so frequently occur, and in the majority of cases it must be admitted that they are due either to lack of fundamental data or to ignorance of the principles involved. If, therefore, what I am about to convey to you in the brief time available appears somewhat elementary, I hope you will realize that we are considering a situation which actually exists and that in many cases the failure of hydroelectric developments to meet expectations has been due to the fact that other elementary principles have been disregarded. I propose to touch only the high spots in regard to estimate of available water supply, including rainfall, run-off, and the effects of storage, and the influence on estimates of cost, of proper exploration, and of the proper evaluation of the uncertain factors.

There are no rainfall or run-off records in this country of sufficient duration to enable an absolute prediction to be made of the available water supply. The best that can be done in any case is an approximation, and the accuracy of this approximation will depend upon the length of record of run-off available together with the contributary evidence on both rainfall and run-off of streams of similar location and characteristics. In cases where no run-off records are available, figures must be based on the rainfall record together with the same contributary evidence. It is possible, however, to determine closely the probable value of the approximate estimate, and both the maximum possible error and the probable error that may be expected from such an approximate estimate. It is also possible from a short-term record to construct a probable long-term record by means of comparison with other long-term records which will give a very much better idea of the

conditions that may be expected than the consideration of a short-term record alone. The method of doing this has been ably outlined by Mr. Allen Hazen in the Transactions of the American Society of Civil Engineers of June, 1914.

It is pertinent to mention at this point the variations that may be expected in rainfall and run-off and the value that should be given to short-term records.

In the average stream in New York State the maximum daily run-off may be one hundred and fifty times the minimum daily run-off. The run-off in an occasional year may be 50 per cent greater or 50 per cent less than the mean annual run-off over a long period of years. When we consider that the distribution may also vary greatly in different years having the same total run-off, we see how dangerous it is to rely only on estimates of output based on mean annual run-off, yet it is my experience that over 50 per cent of engineering reports on hydroelectric projects are based on predictions of output for the average year.

Actual experience shows that the mean determined from a 15-year record of run-off will probably be within 5 per cent of the true mean, but that it may possibly be 13 per cent more or less than the true mean.

These interesting studies on existing records suggest that with a short-term record of any given length the expectations over a long-term period of any desired length can be deduced by the theory of probability. The methods of doing this have been elaborated by Mr. Hazen in his admirable paper and I will not burden you further with them tonight, but I do wish to emphasize the fact that in my belief better results could be obtained by the use of these methods than by reliance on any actual recorded average flow. Furthermore, by the use of these methods a much better estimate can be made of the benefit to be derived by storage, and by an extension of these methods even operating diagrams can be prepared for existing reservoirs which will permit the greatest use of these reservoirs over a long term of years.

Of all the factors which have contributed to the financial failure of hydroelectric projects, none has had a greater influence than underestimating costs. I like to think that if enough money is available there is nothing beyond the power of engineers to accomplish. The hydroelectric project, however, is essentially a commercial project for the purpose of making money, and if it fails in this respect it is a commercial failure. From its very nature, involving as it does work in water, it is hazardous, and it would seem the part of common sense to limit the risk as much as possible by determination in advance of the conditions to be met. Even at the best a large element of risk or uncertainty will remain, due to conditions that cannot be determined in advance or factors that cannot be evaluated. I will pass over these factors of uncertainty which are connected with the construction work such as the organization and management of the construction forces, the amount

and character of the preparatory work that will be necessary, and the actual costs and prices of the various items of the work, with a single caution that prices which apply to one job do not necessarily apply to another in a different part of the country or where conditions are different. This statement may seem elementary, yet how often we see estimates and figures that seem ridiculously low, based on records of actual work done at some other time or place. I am particularly reminded of an estimate of cost of a dam to contain 11,000,000 cu. yd. of earth fill, for which a price was estimated of 11 cents per cu. yd. This was based on an article in the *Engineering-News Record* which stated that earth fill had been placed by hydraulic methods at one of the Reclamation Service dams at a price of 7 cents per cu. yd. No account was taken of difference in conditions and in character of materials. Other things to be guarded against in this connection are the conscious or unconscious unbalancing of quantities and prices and the omission of necessary items.

Assuming that all errors due to assumed cost of work were avoided, the chief reasons for excessive cost for some of our hydroelectric projects have been the failure to determine or interpret properly the underground conditions in advance of construction and erroneous assumptions based upon such lack of knowledge. Up to recent years the hydroelectric project which was properly explored prior to construction has been the exception rather than the rule. In some cases this neglect has been so surprising that it can only be concluded that the builders were afraid to find out in advance what they would have to encounter for fear that the project would fail. In other cases it seems likely that the lack of exploration was due to lack of appreciation by the engineer of the possibility of underground exploration and interpretation, and in some cases even the lack of experience with the problems involved. The most prevailing causes today, however, are the difficulty that engineers have in impressing on the owners the necessity for such thorough exploration, and the difficulty of obtaining the necessary money. It would seem almost axiomatic that where expenditures of millions of dollars were involved, a few thousand dollars for borings and geological investigation would be a reasonable precaution. However, the proper carrying out of the preparatory work on a hydroelectric development is always expensive. It is non-productive work and returns no immediate revenue, and there is usually a constant effort on the part of the owner or developer to cut down expenses. The engineer must have courage enough to insist that sufficient money be allowed for exploration, and he must be able to convince the owner that it will be profitable either in limiting the risk and thereby decreasing the cost or, in some cases, causing the abandonment of an unfavorable project. The trouble is that he has often been instrumental in causing the owner to spend some money on the project and has frequently underestimated the cost of preliminary work. He is between the devil and the deep sea in

recommending further expenditures for borings which by unfavorable results may place him in a position to lose the immediate confidence of the owner.

It should be needless to say that sites of all structures should be carefully explored and that the borings should be deep enough to disclose the true underground conditions. Yet, how often do we see disappointment in construction from borings which penetrate only eight or ten feet into rock, whereas experience shows that a minimum depth of forty feet in rock is none too great. It is needful to call attention to the necessity for careful interpretation of the borings themselves and of the conditions disclosed by them. It is of little use to build a dam if it will not hold water. For the proper interpretation of borings and underground conditions the services of a trained geologist are necessary, as well as for careful and competent inspection of the borings while in progress.

I shall be sorry if in stating these fundamental principles I have in any way indicated pessimism toward hydroelectric development. Successful hydroelectric development furnishes a permanent asset. Capital expenditures properly made in hydroelectric plants return revenue for an indefinite period with minimum operating and replacement charges. It is only by such permanent structures that material progress in civilization is made. For these reasons hydroelectric development is in the long run much preferable to steam even at equal cost. It is a fact, however, that ill-considered and unprofitable hydroelectric developments in the past have made it more difficult to finance meritorious projects at the present time, and it is necessary that we profit by the mistakes of the past in order to build better for the future. I have therefore pointed out some of these fundamental considerations in a rather elementary way in order that you may take them into consideration. We are now in a period of expanding hydroelectric construction and let us realize that failure of individual projects due to neglect of fundamentals has had and will have an unfortunate effect upon the whole.

WATER-POWER COSTS VERSUS STEAM-POWER COSTS

By Geo. A. Orrok [1]

THE kind of a presentation of the subject of water-power development which Mr. Freeman has just favored us with is of great advantage to the profession and more particularly to the people of our country, since the basic principles underlying the development of water power have been pointed out in a clear, logical, and convincing manner for our edification. I have been asked to discuss that portion of the paper covering the question of the cost of water power vs. steam power, and have been particularly asked to cover the general case of large units and transmission

[1] Consulting Engineer, New York, N. Y. Mem. A.S.M.E., A.S.C.E.

from a distance in competition with steam stations of the most modern design situated near the load center.

The design, construction, and operating results of large steam plants are well known, and from examples which have already been constructed we can predicate very accurately the cost figures which most probably would result under any given combination of circumstances. We know, for instance, that it is possible today to construct an electric generating station using steam as

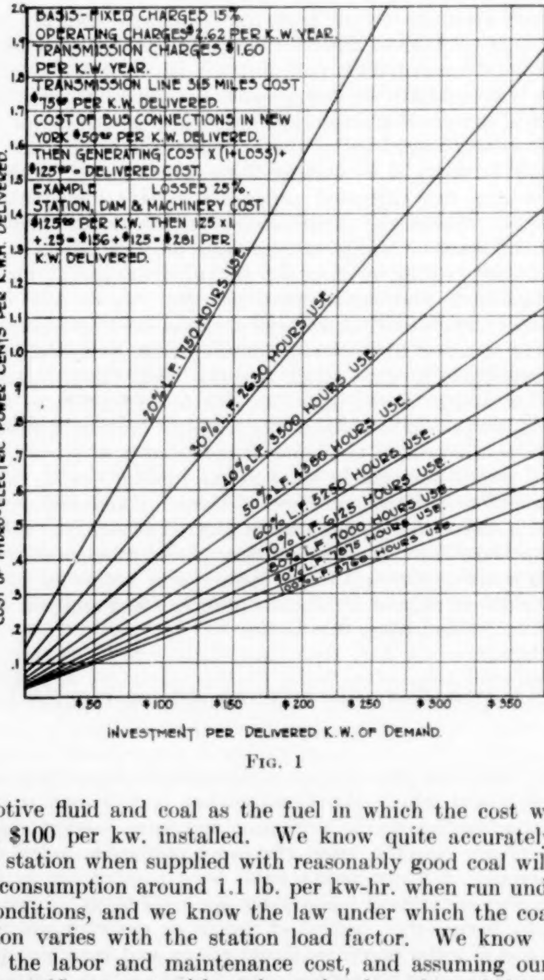

FIG. 1

the motive fluid and coal as the fuel in which the cost will not exceed $100 per kw. installed. We know quite accurately that such a station when supplied with reasonably good coal will have a fuel consumption around 1.1 lb. per kw-hr. when run under the best conditions, and we know the law under which the coal consumption varies with the station load factor. We know within reason the labor and maintenance cost, and assuming our fixed charges at 15 per cent with coal running from $4 to $10 per long ton at the station, suitable curves can be drawn which will show the

cost per unit delivered to the distribution system at any given load factor.

For the steam station I have figured on five 50,000-kw. units, allowing one as reserve and using 200,000 kw. as the peak load on the station from which the load factor has been figured. I have used good coal, Pocahontas or equal, and assumed that the installation will be modern in all its details and located where there is an adequate supply of good condensing water. In general the

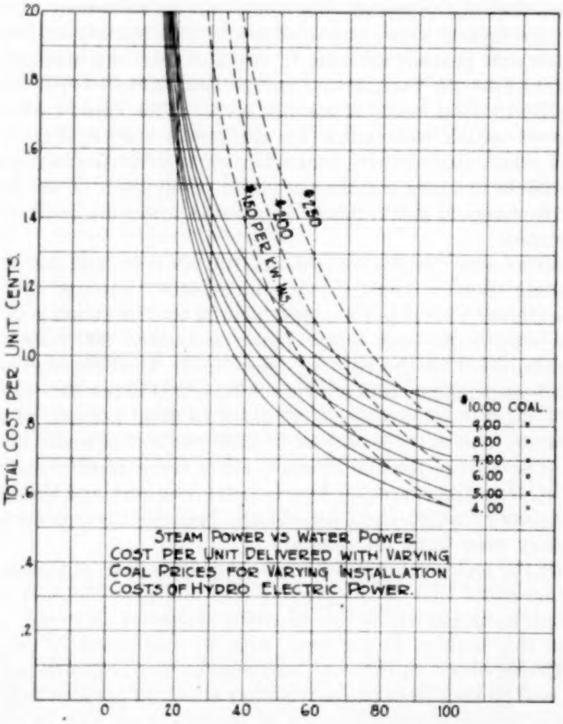

Fig. 2

figures given are such as any good engineering firm would be willing to guarantee to its clients.

With hydroelectric power, on the contrary, no such general statement can be made, and a separate and accurate estimate must be made for each and every condition. Sometime ago in the discussion of a problem of this kind I developed a set of curves, Fig. 1, which covered the cost per unit of electricity generated by water and transmitted over a 300-mile transmission line to the

point of use, which in this instance was a large city, and these curves are sufficiently accurate for our purpose tonight. The transmission line was figured at $50,000 per mile, and the entire plant was proportioned on the basis of a demand on the peak of 200,000 kw. delivered at the point of use 300 miles away. This necessitated 250,000 kw. of installation with no reserve, four circuits, the necessary step-down apparatus, and the high-tension underground cables and other apparatus required to connect up this large amount of power to the distribution system of the city.

As the cost of the installation could only be known in a general way, these figures were made to cover a wide variety of total installation cost per net kilowatt of demand, and the cost per unit was worked out for various load factors from 20 to 100 per cent, or from 1750 to 8760 hours of use per year. With these curves and the curves which were made for the steam station, Fig. 2, the price of which included its connection to the distributing system, it is possible to make a rather accurate comparison of the respective advantages of steam power and water power under the conditions stated.

It will be seen that for $4 coal and 100 per cent load factor, corresponding to an output of one billion, seven hundred and fifty million kilowatt-hours per annum, the total costs of steam generated in the large center and water power generated 300 miles away and transmitted to the city are about even when the cost of the hydraulic generating station does not exceed $150 per kw. installed. This point with $6 coal is about equal to $200 per kw. of water installation, and with $8.50 coal to $250 per kw. of water installation. The curves for the steam station are a good deal flatter, however, than the curves for hydroelectric stations, and the various load factors at which the hydroelectric becomes uneconomical can be readily seen from the diagram.

Using the water-generation curves at $150 per kw. of installation, it will be noted that as the price of coal increases the load factors at which hydroelectric power becomes attractive decrease. With $5 coal this limit is 83 per cent, with $6 coal about 73 per cent, with $7 coal about 64 per cent, and with $8 coal about 58 per cent, but it will be seen from the curves that about 40 per cent is the low limit for the use of hydraulic power, however high may be the coal price.

It is believed that in the calculations which have been made to construct these curves, a leaning has been indicated toward the hydroelectric side rather than to the steam side, and some of my friends have criticized me for using such a high figure for the cost of the steam generating stations. I also have been criticized for not increasing the $50,000 used as the cost of a four-circuit, 250,000-volt transmission line, but I believe that the advantage has been given in this case to the hydroelectric installation.

Some years ago I had occasion to investigate more than three hundred water-power prospects, mostly in this state. Consider-

able work was put into surveys and the estimates showed that the installation cost of these powers averaged $450 per kw. Only a very few ran under $200, and none under $150. It appeared from this investigation that coal must approach $10 to $12 a ton before even a small number of these projects would become commercial. But the larger powers of the St. Lawrence and Niagara may possibly figure in the range indicated in the curves and make up for the additional cost of long-distance transmission.

In prewar times, when coal seldom rose above $5 per ton, $125 per kw. of installation was considered as the limiting sum that could be spent on a hydroelectric installation, even where very short transmission lines were involved. With the raising of the coal price as well as the cost of all construction work, this figure may perhaps be as high as $200 per kw. where the market is not far away.

Mr. Freeman has commented on the fact that the use of hydroelectric power in close proximity to its point of generation brings it usually below the cost of steam, but has also shown that the markets for power are almost invariably situated at a distance from the sources of hydraulic power.

Summing up, the curves show the relation between steam generation at the load center and water generation at a distance with coal prices within the range indicated. The transmission cost does not exceed 20 per cent of the total cost of water power and minor variations in distance or cost affect but little the location of the curves, nevertheless water power, even when more costly than steam power, has a place where coal is costly and may be impossible to obtain. The great extension of hydroelectric stations in certain European countries can only be justified by their lack of coal supplies.

INTERCONNECTION OF POWER SYSTEMS

By Harold W. Buck[1]

THE question of the distribution of power from water powers is a matter inseparably connected with the enterprise.

One of the greatest movements which is taking place today among the great power companies of the country is the extension of their distribution systems over vast territories and of the interconnection between these systems. Those who have not followed in detail the development of power transmission during the past twenty years are perhaps not familiar with how far that tendency has already gone. We hear a great deal nowadays about superpower, and I am afraid that that word is being overworked and is leading the public to believe that some new and mysterious enterprise is coming along the road which is going to cut their power bills in two. That, of course, is not the fact.

[1] Consulting Engineer, New York, N. Y. Fellow A.I.E.E.

Some twenty years ago the Niagara Falls Power Company was engaged in the distribution of 200,000 hp. over a territory of some 10,000 square miles. That had elements of superpower in it, itself. Ten years ago, in the state of California, there was a network of transmission which was practically 700 miles from its northerly limit to its southerly limit. At the present time the entire states of North and South Carolina are connected solidly with a gridiron of lines operated by many water powers and some steam reserve.

Within recent months, I am informed, certain transmission links have been completed so that power can be theoretically transmitted from St. Paul to Chicago, from Chicago to Indianapolis, Indianapolis to Cincinnati, Cincinnati to Louisville, Louisville to Nashville, Nashville to Birmingham, and Birmingham to Atlanta, through the systems of the Northern States Power Company, the Chicago Edison, the Ohio Power, the Kentucky and West Virginia Power Company, the Kentucky Utilities, the Tennessee Power, the Tennessee Eastern Power, the Alabama Power, and the Georgia Railway and Power Company. That is an enormous step in advance in the line of superpower which is already today in existence.

I do not believe, however, that this superpower idea is going to materially reduce the cost of power. I do not see that it is working that way in any respect. It is merely increasing the available supply of power and making it possible for the isolated manufacturer who now operates an efficient plant to connect his works to the network and derive the very great benefit which will result from that.

But the greatest and most important relation which this extension and interconnection of lines has to water power is in making certain water powers available at all. I think it is a great fallacy, this discussion as to which is the cheaper, steam or water power. It befogs the issue and is not the question before us at all. Water power, to be made available, at least in the eastern states of this country, must, I think, in almost every case, be developed and operated in conjunction with a stream plant, and the question before us is: How much of the stream flow of any given river is justified in development, and how much of the proportion must be carried by steam?

At this point Mr. Buck displayed lantern slides showing the hydrographic records of the Tennessee River, the New River, the Hudson River, the Cumberland River, the Feather River, and a river in Japan under minimum- and maximum-flow conditions. He pointed out the necessity for a combination of steam plants with the water power available in order to bring the primary power to a reasonable amount in eastern rivers which flow in territories that have been built up and are occupied by railroads, highways, and other obstructions so that storage is difficult to develop at a reasonable price. For the Feather River, a western stream, ample storage facilities were available, so that steam was not required

in the power development. The Japanese situation was such that the peak load was carried by the water power and a steam plant was operated as a base-load plant at practically 100 per cent load factor.

Mr. Buck concluded his remarks with a plea to those who are interested in water-power development to devote their efforts to the question of the proper combination of steam and water power in order to justify the development of the latter.

GENERAL DISCUSSION

IN DISCUSSION from the floor, William Monroe White [1] quoted an English statesman who said in an open Parliament meeting a few years ago, that the reason why the American manufacturer could compete successfully in the markets of the world and yet pay twice the rate of wage to the American workman that was paid to the British workman, was because each American workman directed twice the horsepower per man that the individual British workman did. Mr. White found that this ratio of available power to wage rate applied also in Japan. He pointed out the small amount of supervision necessary in the operation of hydroelectric plants and compared it with the effort to produce a like amount of power from coal, which requires the uninterrupted service of 1500 men to mine, break, hoist, sort, load, transport, unload, store, rehandle, and fire under the boilers a sufficient amount of coal to produce 40,000 hp., where a hydroelectric plant of the same capacity requires but two men. Mr. White emphasized the fact that the utilizing of fifteen hundred men in this way, employing the already overloaded railways for transporting coal, was a great economic question.

J. P. J. Williams [2] emphasized the interests of the consumer in securing power at the lowest possible rate. He outlined the possibilities of economy in producing hydroelectric power through public ownership and public development and stated that three cents per kilowatt-hour is the price at which the Hydroelectric Power Commission of Ontario is producing power. He stressed the interest of the farmer in cheap power and read the resolutions of the Public Ownership League which advocate complete development and utilization of electric power in the interest of general welfare upon three principles: engineering efficiency, coördination of natural resources with electric utilization, and the interconnection, coördination, and combined operation of a large number of plants or systems of plants.

[1] Manager and Chief Engr. Hydraulic Dept., Allis-Chalmers Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

[2] Civil Eng. Dept., Cooper Union, New York. Mem. A.S.C.E.

T. Kennard Thomson [1] quoted the *Financial Post* of Toronto in a statement that the privately operated companies in the province of Quebec were selling power at a much lower rate than the publicly owned companies of the province of Ontario. Mr. Thomson also called attention to the fact that developing the eight million horsepower from the Niagara and St. Lawrence Rivers during the next ten or fifteen years would only take care of the increased demand for coal or for power supplied by coal in the state of New York alone.

[1] Consulting Engineer, New York. Mem. A.S.M.E.

No. 1911

BOILER TEST RESULTS WITH PREHEATED AIR, COLFAX STATION, DUQUESNE LIGHT COMPANY

By C. W. E. Clarke,[1] New York, N. Y.
Member of the Society

This paper describes the apparatus used and gives the results of an extended series of tests carried out in the summer of 1923 on one of the units of the Colfax Station of the Duquesne Light Company, Pittsburgh, Pa., in which flue-gas air preheaters had been installed. The normal operating rate of evaporating of the boiler employed was the equivalent of 144,000 lb. of water per hour from and at 212 deg. fahr. and the preheater was designed to supply combustion air to the wind box of the stoker at a temperature of 235 deg. fahr. The results obtained show that the efficiency of the unit was increased by from 5 to 7 per cent by the use of the preheater.

THE use of combustion air preheated by flue gas for boiler furnaces is not new either as an idea or an application. The author has for many years felt that this method of heat reclamation if properly designed and installed is the most direct and ideal method. In 1902 he designed and installed preheating equipment in the East St. Louis Plant of Armour & Company. This installation was of elementary design and failed to function properly. It was therefore abandoned.

2 In 1921 the author again began actively to investigate the possibilities of heat reclamation by means of flue-gas air preheaters, proceeding as far as tentative designs of a plate type of preheater for installation in the Colfax Station. Early in 1922 the status of air-preheater results in Europe was investigated. The results actually obtained, even though installations were made under adverse conditions, were such that a trial installation of this type of air preheater was recommended. The equipment was placed in service in 1923.

Description and Arrangement of Apparatus

3 The general arrangement of the preheaters, ducts, dampers, and fan is shown in Fig. 1. The installation was made without any

[1] Power Engineer, Dwight P. Robinson & Co., New York, N. Y.

Contributed by the Power Division and presented at the Annual Meeting, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

addition whatever to the boiler house, the only changes required being an alteration of the uptake and the cutting of holes through the boiler-room floor for the preheated-air duct. The connection between the common forced-draft duct normally supplying the boilers was retained, being cut off by means of a shut-off damper

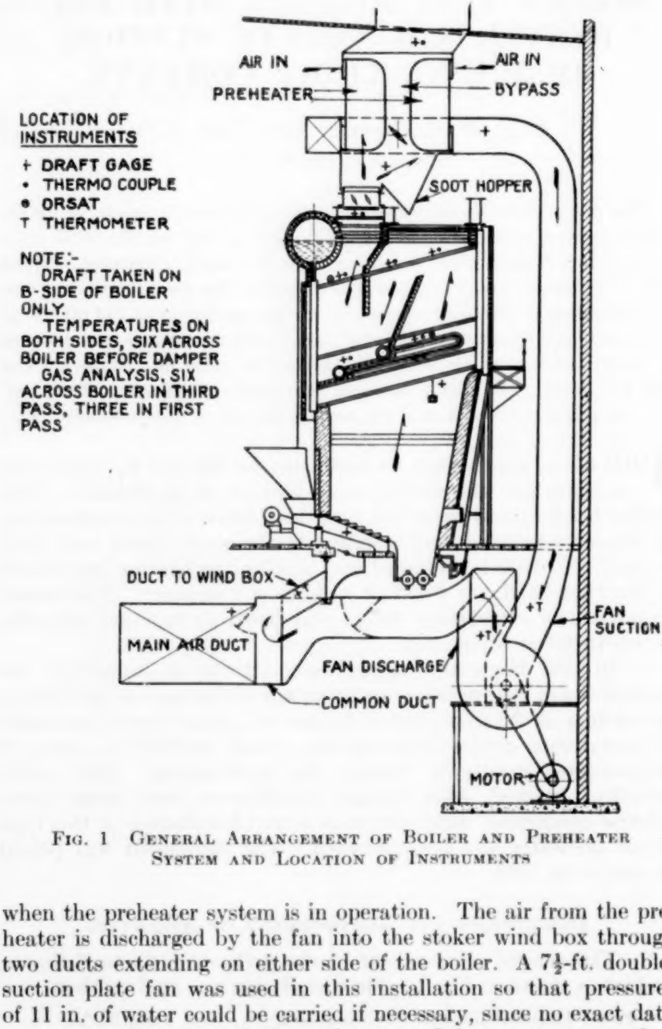

Fig. 1 General Arrangement of Boiler and Preheater System and Location of Instruments

when the preheater system is in operation. The air from the preheater is discharged by the fan into the stoker wind box through two ducts extending on either side of the boiler. A $7\frac{1}{2}$-ft. double-suction plate fan was used in this installation so that pressures of 11 in. of water could be carried if necessary, since no exact data on pressure drop through the preheater and duct system were then available. A belt drive made various fan speeds readily available. This fan is now being replaced by a more compact and efficient

direct-connected motor-driven unit, designed to meet the prevailing conditions. The duct work is insulated to prevent loss of heat.

4 The preheater itself, Fig. 2, was constructed by the Combustion Engineering Corporation, following in general the designs of the Underfeed Stoker Company, Ltd., of London, England, except that the heating elements are placed back to back and separated

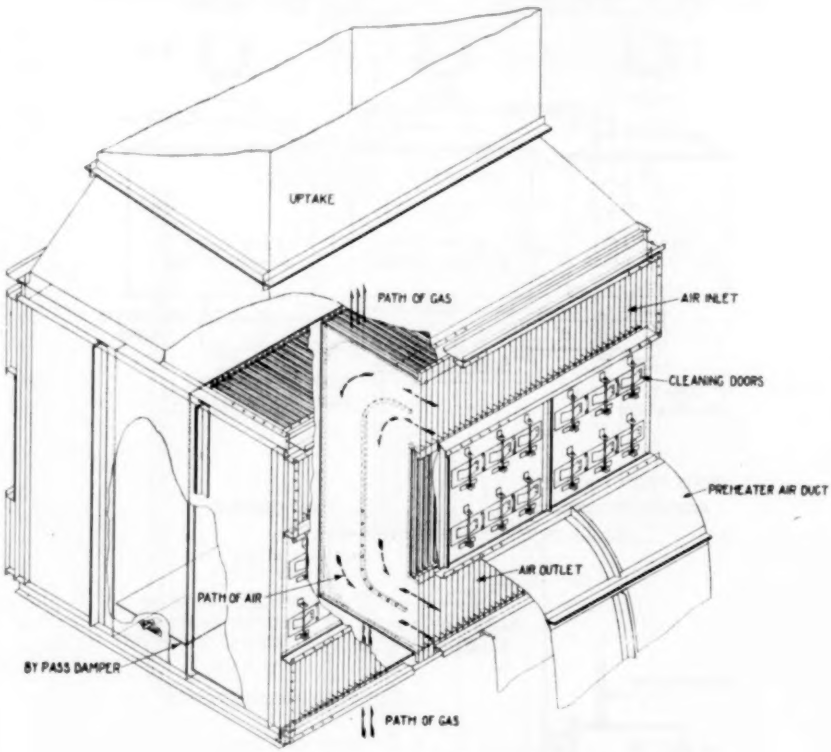

Fig. 2 Detail of Preheater

by a by-pass space fitted with a damper. The preheater is designed for operation when the boiler is evaporating the equivalent of 144,000 lb. of water per hour from and at 212 deg. fahr., which is the normal operating rate of the Colfax boilers. It was not considered worth while to install preheating equipment for ratings higher than this as extreme ratings are emergency conditions and the savings obtained would not justify the additional expenditure. When high overload ratings are required the preheater by-pass damper is opened to allow more uptake area.

5 The arrangement of two banks of elements back to back

renders the maximum surface available in a given vertical dimension. The preheater was guaranteed to supply air to the wind box at a temperature of 235 deg. fahr., the boiler operating at the evaporation rate given above. The tests indicate that the equivalent of this was actually obtained.

6 The preheater is riveted and calked, but it was found im-

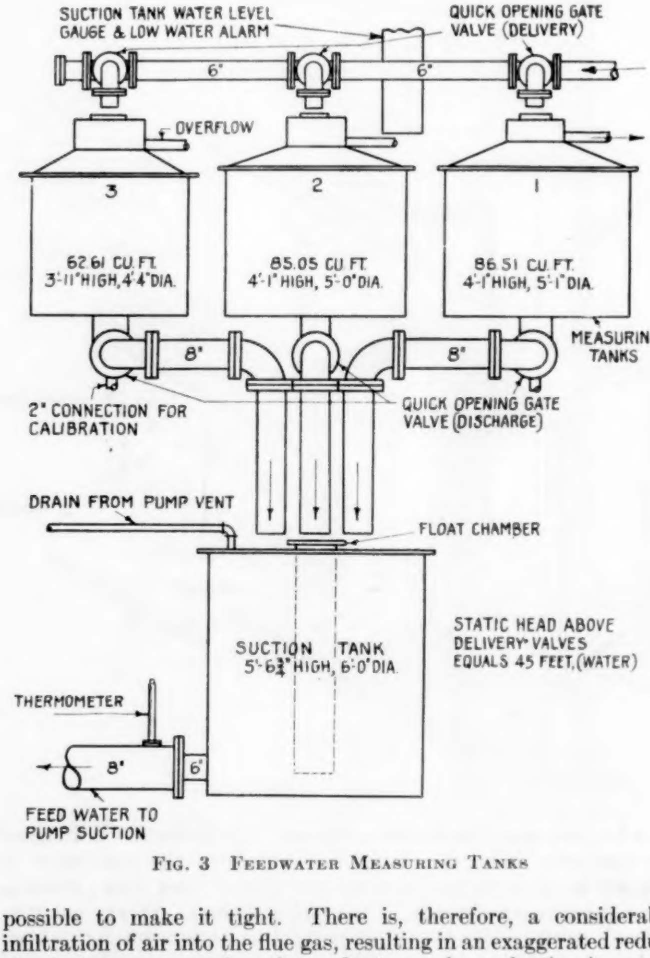

Fig. 3 Feedwater Measuring Tanks

possible to make it tight. There is, therefore, a considerable infiltration of air into the flue gas, resulting in an exaggerated reduction in temperature after the preheater and a reduction in actual preheater performance. A new design using welded construction throughout has been developed and will be used for the installations now under way. An increase in preheater efficiency should be gained thereby.

7 The tests reported in this paper were made to determine the increase in efficiency of the boiler unit when using preheating equipment over and above that obtained without such equipment. Certain local conditions are inherent in the arrangement of apparatus for boiler No. 9 at Colfax which may not obtain for a similar installation elsewhere. One point is that the temperature of the inlet air to the preheater is somewhat high, due to the preheater suction being located directly above the boiler tested. It is desired

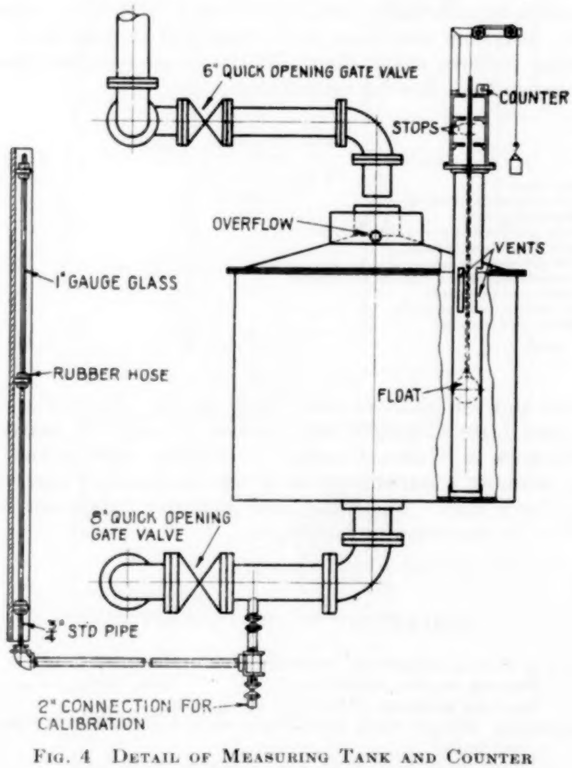

Fig. 4 Detail of Measuring Tank and Counter

to present here actual results, rather than calculations which would bring these conditions to a common basis.

8 Tests were run on boiler No. 9 at the Colfax Station from July 14 to August 4. The runs were each of 24 hours' duration. Each series, one with and one without the preheater, was continuous for periods of eight and thirteen days, respectively. No attempt was made to establish high efficiencies. Throughout the test the boiler and stoker were handled by the regular boiler-room force, with some assistance from the Westinghouse stoker engineers.

The operation did not differ from the regular daily method, with the exception of the manipulation of the wind-box dampers to give as nearly a constant rating as possible during the period of each test. In short, every effort was made to have the test results representative of day-to-day operation.

9 The tests were directed and supervised by Dwight P. Robinson & Company, Incorporated, assisted by representatives of the Duquesne Light Company, the Babcock & Wilcox Company and the Westinghouse Company. They were conducted as far as possible in accordance with the A.S.M.E. test code. Approximately forty-four observers were employed during these tests, operating in three shifts, their situations and functions being as indicated in the following tabulation:

Position	No. of Men Per Shift	Total
Test chief	1	3
Coal weights and samples	1	3
Measuring water	3	9
Main gage board and general data	1	3
Data at preheater fan	1	3
Flue-gas analysis	3	9
Thermocouples and optical pyrometer	1	3
Drafts and pressures	1	3
Ash weights and samples	2	2
Extras and records	2	6
Total	16	44

All coal and ash analyses were made by the laboratories of the Duquesne Light Company and checked by separate analyses by the Babcock & Wilcox Company. Readings were taken every fifteen minutes, a signal horn being used to start all observations at the same time. The boiler and preheater equipment is best described by the following tabulation:

BOILER NO. 9 COLFAX STATION

Babcock & Wilcox cross-drum, water-tube boiler, installed in 1922.
Heating surface, 22,914 sq. ft.
Working pressure, 275 lb. per sq. in. gage.
Normal rating (100 per cent), 79,053 lb. of water from and at 212 deg. fahr. per hour.
Tubes, 20 high, 51 wide, 4 in. diameter, 20 ft. long.
Drum, 60 in. diameter, 34 ft. 1 in. long.
Setting, 30 ft. 1 in. wide inside of walls.
Superheater, Babcock & Wilcox, containing 2999 sq. ft. of heating surface, located in first pass of boiler.
Furnace volume, 7500 cu. ft.
Stoker, Westinghouse underfeed, 17-retort, 20-tuyere, extra long with side-wall tuyeres.
Clinker grinder, double-roll, separate drive on each side of boiler:
Grate, 29 ft. 10-1/4 in. wide by 13 ft. 6 in. effective length.
Projected area, 218.31 sq. ft.
Grinder-pit section, 163.89 sq. ft.
Preheating surface, 11,200 sq. ft.

10 Feedwater was measured in the tanks shown in Fig. 3. The upper three tanks, used to measure the water, emptied into the lower receiving tank connected to the pump suction. The measuring tanks were filled to the top of the small neck. Excess water overflowed into the surrounding sleeve which was drained to the condensate system. The small section at the top of the measuring tank of course insures the maximum of accuracy in

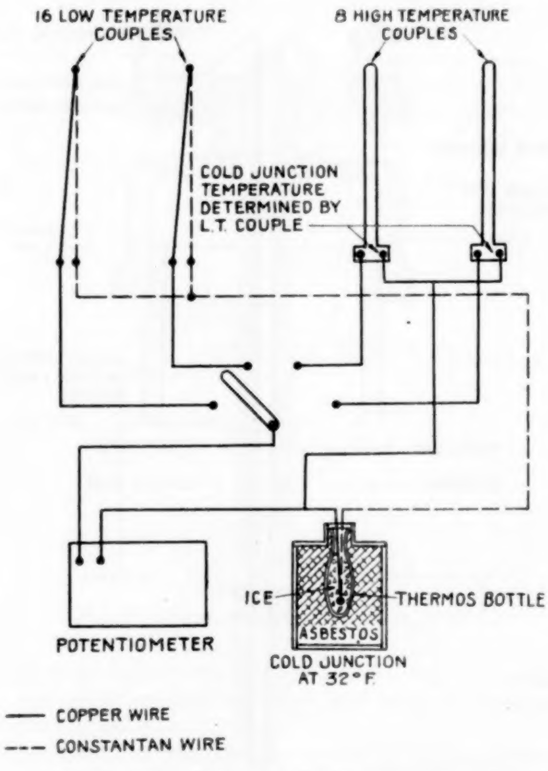

Fig. 5 Wiring Scheme for Thermocouples

water measurement. It was found that two tanks were sufficient to handle the water at a maximum boiler rating of 150,000 lb. per hour. The third tank was used as a spare.

11 Bleeder connections were placed in connecting pipe lines wherever there was any danger of leakage and were kept open during the tests. Blow-down lines were checked up by touching once a day. Counters placed on the measuring tanks proved satisfactory in checking the record kept by the observers. See Fig. 4.

12 Coal was weighed on the regular station larry, the weights being checked by an observer who also collected samples for analysis. A counter for stoker-ram strokes was placed on each of the seven sections of the stoker as an additional check on the coal fired.

13 Samples for analysis were taken equally from each section

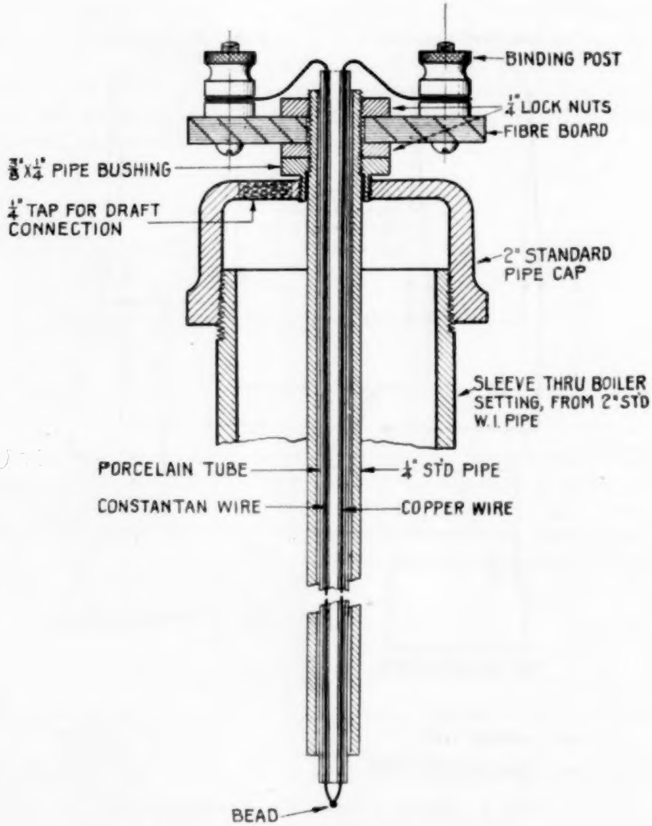

Fig. 6 Detail of Low-Temperature Thermocouples

of the stoker hopper across the boiler immediately after the hopper was filled. One per cent of the coal fired was collected in this manner and worked down to a laboratory sample every eight hours. A special sample was taken for moisture determination. This sample was kept tightly covered in a cool part of boiler room.

14 The ash was weighed wet and a sample taken for moisture, which sample was dried before quartering for the laboratory. A

sample for determination of combustible in ash was taken by dumping all the refuse on a wooden platform and saving one out of every ten shovelfuls. The part saved was then crushed and quartered down for the laboratory sample.

15 Fuel-bed and front-wall temperatures were determined by means of an optical pyrometer.

16 It was originally intended to sight upon a target suspended from the first row of tubes, Fig. 1, to determine the gas temperatures at the entrance to tubes, but it was found impossible to keep the brick in place longer than two days. However, the front-wall temperature, as seen from the same observation holes, corresponded very closely to the brick temperature, so that these were taken after the brick disappeared. The radiation effect from the

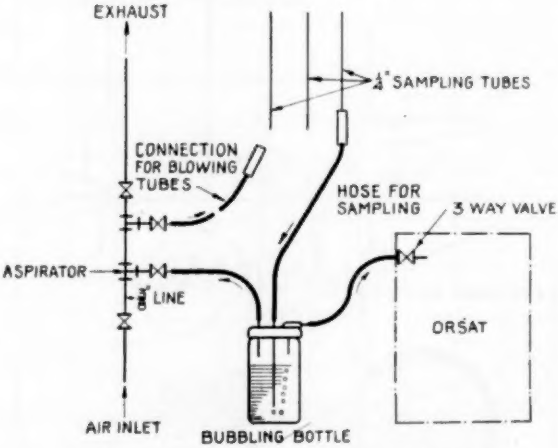

Fig. 7 Connections for Gas Sampling

target and front wall to the relatively cool tubes is considerable, but for comparing temperatures with and without preheated air this error is negligible.

17 Thermocouples were used to determine the gas temperature through the boiler. The low-temperature couples were made of No. 22 gage copper-constantan wire, the fine wire with the small bead reducing the radiation effect to a minimum. Special rare-metal couples were used for the higher temperatures, these being enclosed in wrought-iron pipe or quartz tubes, depending on the temperature to which they were subjected. The radiation effect on these couples was probably greater than in the case of the low-temperature couples, ,but the error should not affect the comparison of tests with and without preheater. The couples were 6 ft. overall and extended to a point approximately 4 ft. inside the furnace wall. The position of the couples is indicated in Fig. 1,

the wiring scheme in Fig. 5, and the detail of the low-temperature couples in Fig. 6. Temperatures in the preheated-air ducts were taken by means of indicating thermometers inserted in an oil well extending entirely across the duct.

18 Flue-gas samples were taken at six points equally spaced

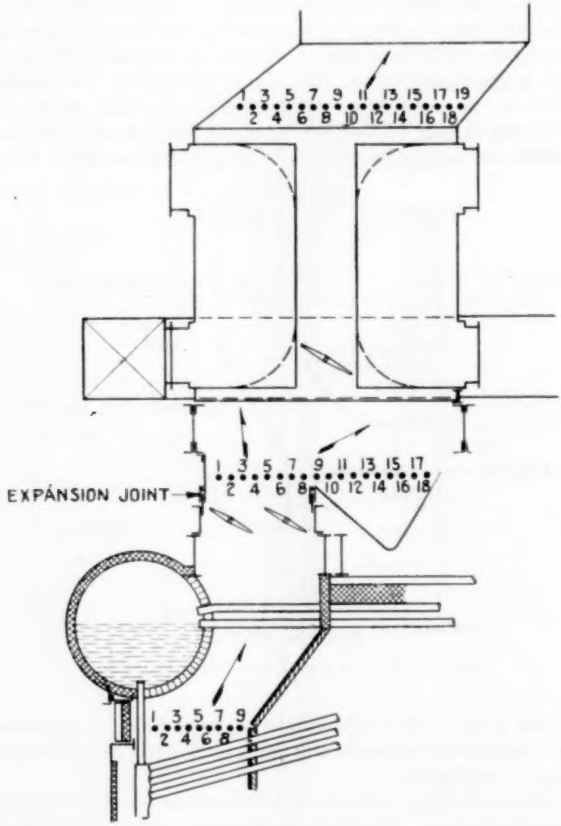

Fig. 8 Location of CO_2 Traverses at Preheater

across the boiler at the top of the third pass, and at three points in the center of the first pass, also equally spaced as indicated in Fig. 1. Fig. 7 shows the arrangement of apparatus for sampling. Three Orsats were used continually: two in the third pass and one in the first pass. Analyses were made in rotation, gases at each point being analyzed once every 45 minutes. It was found unnecessary to use water-cooled sampling pipes in the first pass on account of the relatively low temperatures.

19 Special sets of CO_2 traverses were taken at the center of the boiler at the top of the third pass from front to back, and at the center of the uptake before and after preheater from front to back as indicated in Fig. 8. One set of the three traverses was made for each 24-hour test run with the preheater in operation.

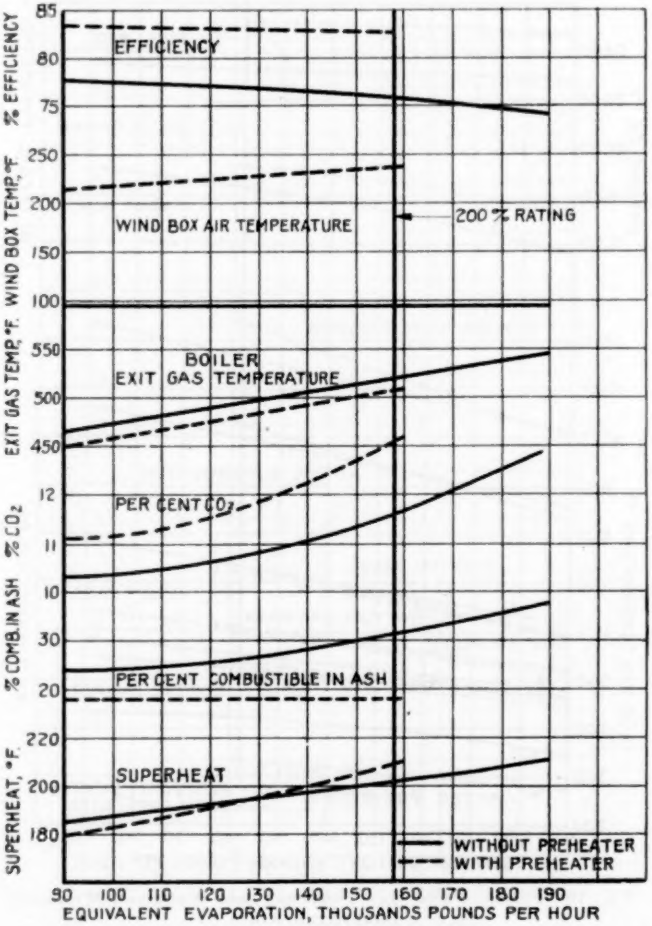

Fig. 9 Comparative Performance of Boiler No. 9 with and without Preheated Air

20 Drafts were taken at the same points as the gas temperatures, connections being made to a common 2-in. sleeve through the boiler setting. Pipe lines were run from the draft connections to a common draft-gage board with gages arranged in the same order as draft points in the boiler.

21 Scales were used to determine weight of coal, weight of total refuse (wet), of sample of refuse (dried) and for the calibration of the water-measuring tanks. Each was calibrated sometime during each test. All were found to be accurate with the exception of a

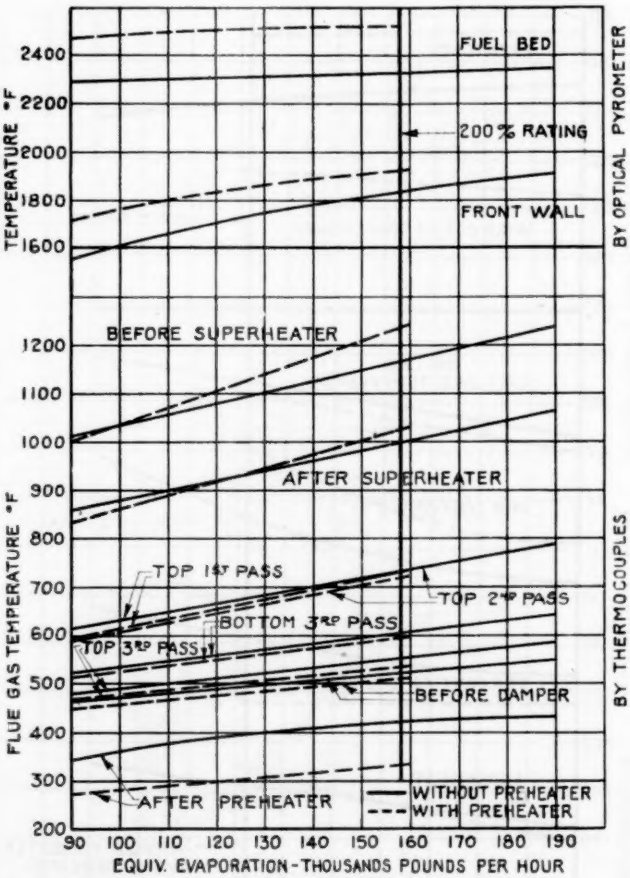

Fig. 10 Furnace and Gas Temperatures With and Without Preheated Air

small error in the ash-weighing scales, which were calibrated and the results corrected.

22 All thermometers were calibrated during the test and the more important temperatures corrected accordingly. Thermometers calibrated for 3 in. immersion were used in practically all cases.

Comments on Test Data and Results

23 Referring to the data and results of the evaporative test on No. 9 boiler unit with and without air preheater, given in Tables 1 and 2, the following explanations and comments seem advisable:

24 In the test runs shown in Table 2, tests with preheater begin

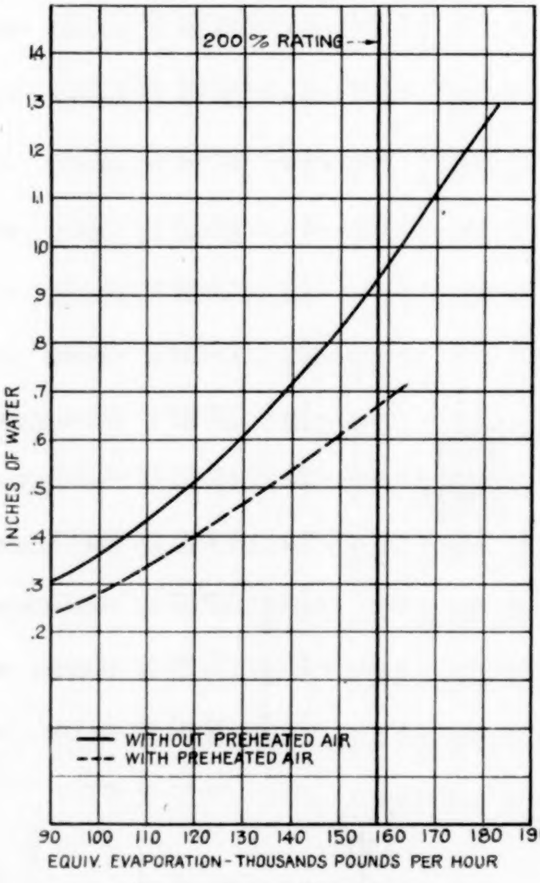

Fig. 11 Boiler Draft Loss

with test No. 4. Three preliminary tests were run, but due to trouble with the preheater fan these were not considered official.

25 Gas temperatures above the preheater, shown under Item 14 (*a*) with the preheater in operation, are unduly low because of air infiltration through the preheater.

26 The flue-gas temperature before the damper, Item 14 (*b*),

Table 1 Data and Results of Evaporative Test (without Preheater)

DUQUESNE LIGHT COMPANY
COLFAX POWER STATION
DWIGHT P. ROBINSON & COMPANY, INC.,
SPRINGDALE, PENNA.
DATA AND RESULTS OF EVAPORATIVE TEST
NO. 9 BOILER UNIT
(WITHOUT PREHEATER)

B. & W. Boiler -
Water Heating Surface - 22,914 Sq.Ft.
B. & W. Superheater -
Superheating Surface - 2,990 "
Westinghouse, 17 Retort, 20 Tuyere, Underfeed Stoker -
Underfeed Section (Projected Area) - 216.21 "
Grinder Pit Section - 163.86 "

SERIES 1000 - TEST NO.		1	2	3	4	5	6	7	8	9	10	11	12	13	1 to 13 Incl.
8 - Date Started (4:00 P.M.)		7/23/23	7/24/23	7/25/23	7/26/23	7/27/23	7/28/23	7/29/23	7/30/23	7/31/23	8/1/23	8/2/23	8/3/23	8/4/23	7/23 to 8/5, '23
9 - Duration (Hours)		24	24	24	24	24	24	24	24	24	24	24	24	24	312
10 - Kind and Size of Coal		HARWICK RUN OF MINE - BITUMINOUS													
AVERAGE PRESSURES AND TEMPERATURES															
11 - Steam Pressure Superheater Outlet, #/Sq."	Ga.	260.3	260.0	268.5	267.9	267.5	267.2	267.2	262.9	265.6	268.7	268.4	265.0	261.2	266.3
(a) Steam Pressure Drum	"	278.0	277.1	277.2	275.2	275.3	274.7	275.5	277.6	275.9	272.8	274.0	272.2	266.5	275.1
(b) Barometric Pressure	"Hg.	29.24	29.09	29.24	29.20	29.16	29.06	29.16	29.09	29.21	29.27	29.24	29.21	29.21	29.21
12 - Steam Temperature at Superheater Outlet -	°F	614.9	612.2	600.0	605.7	603.1	598.3	604.7	611.6	615.3	614.0	618.2	614.1	595.3	608.9
(a) Corresponding Saturation Temperature	"	412.0	412.1	412.1	411.9	411.8	411.7	411.7	412.2	411.3	410.6	410.5	410.3	409.7	411.4
(b) Superheat	"	202.9	200.1	190.9	193.8	191.3	186.6	193.0	199.4	204.1	203.4	207.7	203.8	185.6	197.5
13 - Feed Water Temperature at Boiler -	°F	120.3	120.4	118.2	121.4	118.2	123.2	122.1	118.8	116.5	116.5	115.8	117.2	127.8	119.8
(a) Heat in Steam above Feed Water Temperature	B.t.u./#	1230.8	1234.2	1239.3	1224.9	1224.9	1219.4	1223.8	1233.4	1234.7	1238.2	1237.3	1230.6	1218.1	1229.2
14 - Average Flue Gas Temperatures -															
(a) After Preheater	°F	427	413	390	386	398	362	400	407	440	401	432	421	386	408
(b) Before Damper (Thermocouples)	"	[illegible]	[illegible]	518	489	490	475	480	[illegible]	[illegible]	[illegible]	536	516	[illegible]	[illegible]
(c) Top 3rd Pass	"	[illegible]	[illegible]	[illegible]	[illegible]	519	498	506	[illegible]	553	501	566	542	[illegible]	534
(d) Bottom " "	"	603	628	[illegible]	589	542	530	554	[illegible]	609	610	666	601	[illegible]	597
(e) Top 2nd "	"	772	792	752	704	641	622	659	696	732	737	759	717	618	708
(f) " 1st "	"	774	762	784	706	648	624	671	685	726	798	755	718	636	704
(g) After Superheater	"	1040	1090	1007	970	949	864	923	953	1011	996	1019	995	879	971
(h) Before "	"	1173	1196	1198	1164	1191	1015	1090	1155	1207	1194	1172	1141	1089	1189
(i) Temperature of Front Wall by Optical Pyrometer	"		2061	2026	1940	1997	1815								
(j) " " Fuel Bed " " "	"			2536	2497	2508	2495								
(k) " " Bridge Wall " " "	"			2135	2063	2006	1988								
(l) Average Temperature of Boiler Casing	"	118	148	154	159	157	153	150	[illegible]	158	170	148	161	164	154
(m) Temperature before Damper (Foxboro)	"	570	[illegible]	541	[illegible]	514	487	518	507	552	545	548	546	498	538
(n) " Top 3rd Pass (Bailey)	"	540	503	518	508	465	448	468	[illegible]	504	506	536	514	450	527

TABLE 1 (*Continued*)

SERIES 1002 - TEST NO.		1	2	3	4	5	6	7	8	9	10	11	12	13	1 to 13 Incl.
16 - Drafts by Manometer															
(a) Over Fire	" Water	Flue .08	.06	.06	.03	.08	.04	.02	.02	.02	.01	.08	.04	.02	.03
(b) Before Superheater	"	.01	.10	.08	.04	.06	.03	.02	.04	.05	.24	.07	.07	.01	.05
(c) After "	"	.31	.09	.11	.06	.06	.03	.05	.06	.36	.08	.06	.08	Flue .01	.05
(d) Top 1st Pass	"	.19	.27	.21	.15	.08	.05	.07	.13	.10	.10	.26	.13	.30	.15
(e) " 2nd "	"	.33	.28	.22	.17	.12	.07	.08	.13	.30	.80	.38	.21	.01	.17
(f) Bottom " "	"	.36	.58	.52	.44	.33	.22	.29	.42	.48	.41	.62	.60	.17	.44
(g) Top 3rd "	"	.77	1.02	.82	.71	.49	.34	.46	.46	.73	.97	1.01	.58	.38	.71
(h) Before Damper	"	1.11	1.17	.93	.61	.58	.43	.54	.41	.21	.96	1.30	.95	.33	.83
(i) " Economizer	"	1.16	1.20	1.25	1.32	1.38	1.24	1.25	1.13	1.26	1.22	1.18	1.11	1.23	1.20
(j) After "	"	1.30	1.19	1.21	1.19	1.17	1.15	1.10	1.32	1.21	1.29	1.10	1.15	1.15	1.10
(k) At Base of Stack	"	1.46	1.46	1.43	1.43	1.38	1.29	1.31	1.39	1.42	1.39	1.40	1.44	1.18	1.36
(l) " Fan Suction	"						.81	.99	1.59	1.59	1.23				
(m) Pressure at Fan Discharge	"						7.02	6.55	6.88	6.32	6.24				
(n) " Forced Draft Duct	"	7.59	6.89	5.70	5.81	4.79	6.51	5.13	6.38	5.32	5.37	7.16	6.07	5.13	5.76
(o) " Duct to Wind Box	"	7.40	6.95	5.70	5.78	4.97	7.31	7.21	6.88	6.60	6.47	7.17	6.08	5.30	6.44
(p) " Under Grate	"	6.58	5.77	5.12	4.78	3.96	3.04	4.08	6.65	5.33	5.40	6.10	5.28	5.00	6.85
(q) " Bridge Wall Air Box	"	3.09	1.54	.18	.00	.01	.00	.01	.18	.72	.90	2.19	.11	.06	.66
18 - State of Weather -		Rain	Cloudy	Clear	Clear	Rain	Clear	Clear	Cloudy	Clear	Clear	Rain	Clear	Cloudy	
(a) Temperature of External Air	°F	68.4	71.0	68.2	68.4	72.8	75.0	74.2	71.8	72.4	78.2	76.6	74.2		72.4
(b) " " Air at Main Gauge Board	"	76.3	79.7	76.4	77.6	90.8	80.8	80.7	81.1	81.8	85.2	85.5	83.7	86.0	81.1
(c) " " " " Draft " "	"	86.3	90.6	87.1	88.0	93.0	92.3	93.8	98.3	94.2	97.0	98.6	96.1	90.2	93.0
(d) " " " In Main Air Duct	"		106.0	101.0	100.0	109.8			107.0	108.0	110.8	115.0	120.0		
(e) " " " " Wind Box	"	97.4	97.8	91.7	91.6	90.8	102.7	94.6	87.0	87.4	91.0	103.7	100.9	108.2	95.8
(f) Relative Humidity	%	67.5	69.0	54.5	75.5	76.8	80.0	60.8	62.8	59.0	68.0	70.0	61.0	68.0	66.0
(g) Dry Bulb Temperature	°F	79.4	89.3	62.7	89.4	87.0	83.3	94.5	94.8	80.7	84.3	82.6	94.4	90.0	87.3
(h) Temperature of Coal	"	72.3	70.0	71.0	69.0	70.1	71.6	72.2	76.8	75.2	76.7	76.2	79.0	75.8	75.1
TOTAL QUANTITIES															
19 - Total Weight of Coal as Fired -	lbs.	469,975	412,930	384,238	331,000	282,160	231,046	289,568	338,080	391,488	388,240	449,428	359,690	224,040	4,501,370
(a) Average Revolutions of Stoker Sections		1,208	1,199	965	929	791	641	791	928	1,115	1,093	1,230	1,002	468	12,608
(b) Total Ram Strokes		22,238	19,703	16,711	15,790	13,447	10,890	13,440	16,864	18,974	18,806	20,935	17,034	10,681	214,367
(c) Weight of Dry Coal Per Stroke	lbs.	19.87	20.12	20.42	20.45	20.14	20.30	20.58	20.27	19.86	20.11	20.62	20.34	20.07	20.22
20 - Moisture in Coal as Fired -	%	5.74	5.98	5.67	2.48	4.00	3.86	4.07	3.72	3.76	3.24	3.69	3.46	4.31	3.70
21 - Total Weight of Dry Coal -	lbs.	441,609	388,476	341,284	322,902	270,854	222,080	277,938	321,886	376,763	375,748	432,851	346,588	214,384	4,335,140
22(a) Ash and Refuse (Weighed) -															
(1) Total Weight Wet	lbs.	85,100	82,830	66,190	59,865	49,080	35,380	46,220	67,170	79,060	59,850	108,840	63,140	42,170	850,465
(2) Moisture by Analysis	%	24.19	26.72	22.35	24.84	27.07	22.82	20.44	28.87	25.06	26.98	29.98	29.47	23.73	24.83
(3) Ash and Refuse Based on Dry Coal	"	14.36	15.31	12.50	13.81	13.38	12.48	12.07	14.88	15.80	17.48	17.08	12.85	15.00	14.85
(4) Total Weight Dry	lbs.	62,056	60,668	48,250	44,607	35,796	28,191	33,069	47,779	58,756	43,815	73,988	44,536	32,163	630,460
(b) Ash and Refuse (From Analysis) -															
(1) Ash and Refuse Based on Dry Coal	%	17.43	19.17	18.94	18.50	15.26	16.45	14.83	18.88	18.09	18.77	21.78	19.63	14.90	18.01
(2) Total Weight Dry	lbs.	77,007	74,064	64,304	59,699	41,332	36,734	38,981	59,643	68,702	70,182	94,146	68,716	32,378	780,112
23 - Total Combustible Burned (From 22-b) -	lbs.	364,826	312,679	286,978	268,026	229,519	187,316	234,980	261,904	307,056	305,606	338,719	277,798	179,017	3,565,028
24 - Sprinkler Water to Ash Pit -															
(a) Total Revolutions of Clinker Grinder Rolls		52	48	44	47	38	34	31	43	40	34	43	38	33	504

TABLE 1 (*Continued*) DATA AND RESULTS OF EVAPORATIVE TEST (WITHOUT PREHEATER)

SERIES 1002 - TEST NO.		1	2	3	4	5	6	7
25 - Total Weight of Water -								
(a) By Tank Measurement	lbs.	3,569,301	3,237,376	2,894,540	2,642,113	2,287,422	1,890,216	2,318,080
(b) " Venturi Meter Integrators	"		3,204,500	2,890,300	2,656,300	2,300,400	1,907,100	2,332,900
(c) Difference from Tank Measurement	%		Minus 1.02	Plus .16	Plus .54	Plus .57	Plus .89	Plus .64
(d) By Venturi Meter Planimetered Charts	lbs.	3,614,300	3,271,100	2,937,100	2,710,600	2,365,900	1,970,300	2,421,000
(e) Difference from Tank Measurement	%	Plus1.26	Plus 1.04	Plus 1.47	Plus 2.60	Plus 3.36	Plus 4.23	Plus 4.44
(f) By Bailey Meter Integrators	lbs.	3,625,400	3,283,400	2,951,500	2,670,000	2,314,600	1,896,500	2,330,000
(g) Difference from Tank Measurement	%	Plus 1.57	Plus 1.42	Plus 1.97	Plus 1.08	Plus 1.19	Plus .35	Plus .58
26 - Total Water Evaporated -	lbs.	3,569,301	3,237,376	2,894,560	2,642,113	2,287,422	1,890,216	2,318,020
27 - Factor of Evaporation -		1.2684	1.2686	1.2660	1.2625	1.2643	1.2526	1.2611
28 - Total Equivalent Evaporation From and At 212 °F	lbs.	4,527,301	4,090,740	3,664,520	3,336,150	2,891,987	2,378,246	2,923,256
HOURLY QUANTITIES AND RATES								
29 - Dry Coal Per Hour -	lbs.	16,409	16,820	14,219	13,435	11,306	9,253	11,414
(a) Dry Coal Per Retort Per Hour	"	1082.9	972.0	836.4	791.4	663.9	544.0	671.4
30 - Dry Coal Per Sq.Ft of Grate Surface, Per Hour -	"	84.33	75.67	65.13	61.60	51.70	42.30	52.29
31 - Water Evaporated Per Hour -	"	148,731	136,891	120,607	110,088	94,943	78,759	96,004
32 - Equivalent Evaporation From and At 212 °F Per Hour -	"	188,630	170,447	152,765	138,964	120,500	98,369	121,802
33 - " " Per Sq.Ft. of Heating Surface, Per Hour -	"	8.33	7.46	6.67	6.06	5.26	4.32	5.32
34 - Dry Flue Gases Per Hour -	"	288,492	236,364	217,693	214,380	190,735	160,910	168,790
CAPACITY								
36 - Percent of Rating -	%	238.7	215.6	193.3	175.6	152.6	125.3	154.1
ECONOMY								
37 - Water Evaporated Per Lb. Coal as Fired -	lbs.	7.7770	7.8404	8.1730	7.9680	8.1076	8.1012	8.1176
38 - " " " " Dry Coal -	"	8.0786	8.1604	8.4681	8.1873	8.4462	8.5116	8.4680
39 - Equivalent Evaporation Per Lb. Coal as Fired -	"	9.8639	9.9071	10.3806	10.0616	10.2502	10.2006	10.2371
40 - " " " " Dry Coal -	"	10.2472	10.3179	10.7452	10.3364	10.6778	10.6664	10.6714
41 - " " " " Combustible -	"	12.6095	12.7645	12.7621	12.6609	12.6002	12.6904	12.4420
42 - Dry Flue Gas Per Lb. Dry Coal -	"	13.77	14.20	15.21	15.93	16.90	17.39	16.01
EFFICIENCY								
43 - B.T.U. Per Lb. Coal as Fired, by Calorimeter -	B.t.u.	12,717	12,762	12,701	12,761	12,935	12,682	12,941
(a) B.T.U. Per Lb. Dry Coal	"	13,215	13,291	13,278	13,112	13,476	13,351	13,490
(b) " " " Combustible	"	16,140	16,185	16,137	16,208	16,184	16,178	16,194
44 - Efficiency of Unit -	%	74.69	75.35	76.53	76.44	76.90	77.74	76.76
45 - " " " Based on Combustible -	"	79.40	81.89	81.94	80.91	80.88	81.97	79.46

SERIES 1002 - TEST NO.		8	9	10	11	12	13	1 to 13 Incl.
25 - Total Weight of Water -								
(a) By Tank Measurement	lbs.	2,620,136	3,063,384	2,991,150	3,304,660	2,641,801	1,850,666	35,510,842
(b) " Venturi Meter Integrators	"	2,636,900	3,065,700	2,991,600	3,274,600	2,836,100	1,859,800	
(c) Difference from Tank Measurement	%	Plus .64	Plus .08	Plus .01	Minus .91	Minus .20	Plus .49	Plus .07
(d) By Venturi Meter Planimetered Charts	lbs.	2,729,200	3,124,500	3,060,000	3,334,600	2,875,700	1,906,600	36,328,000
(e) Difference from Tank Measurement	%	Plus 4.16	Plus 2.00	Plus 2.26	Plus .91	Plus 1.19	Plus 3.02	Plus 2.30
(f) By Bailey Meter Integrators	lbs.	2,656,000	3,090,000	3,035,300	3,348,100	2,880,600	1,858,100	35,930,000
(g) Difference from Tank Measurement	%	Plus 1.37	Plus .85	Plus 1.40	Plus 1.34	Plus 1.37	Plus .40	Plus 1.20
26 - Total Water Evaporated -	lbs.	2,620,136	3,063,384	2,991,150	3,304,660	2,641,801	1,850,666	35,510,843
27 - Factor of Evaporation -		1.2712	1.2724	1.2686	1.2780	1.2712	1.2532	1.2467
28 - Total Equivalent Evaporation From and At 212 °F	lbs.	3,330,717	3,897,727	3,796,174	4,213,465	3,612,497	2,317,420	44,360,501
HOURLY QUANTITIES AND RATES								
29 - Dry Coal Per Hour -	lbs.	13,098	15,690	15,073	18,058	14,429	9,938	13,095
(a) Dry Coal Per Retort Per Hour	"	780.1	923.5	914.1	1060.9	848.4	525.5	817.4
30 - Dry Coal Per Sq.Ft of Grate Surface, Per Hour -	"	61.37	71.91	71.38	83.61	66.14	40.32	63.65
31 - Water Evaporated Per Hour -	"	109,172	127,641	124,630	137,695	118,404	77,112	113,765
32 - Equivalent Evaporation From and At 212 °F Per Hour -	"	138,780	162,405	158,257	175,561	150,521	96,560	144,168
33 - " " Per Sq.Ft. of Heating Surface, Per Hour -	"	6.06	7.10	6.91	7.66	6.57	4.21	6.30
34 - Dry Flue Gases Per Hour -	"	206,989	226,175	227,820	242,473	216,286	146,819	209,676
CAPACITY								
36 - Percent of Rating -	%	175.6	205.5	200.3	222.1	190.4	122.8	182.4
ECONOMY								
37 - Water Evaporated Per Lb. Coal as Fired -	lbs.	7.8468	7.6290	7.7430	7.3329	7.9007	8.2808	7.8865
38 - " " " " Dry Coal -	"	8.1463	8.1306	8.0082	7.6347	8.2009	8.6326	8.1914
39 - Equivalent Evaporation Per Lb. Coal as Fired -	"	9.9728	9.9565	9.8232	9.3750	10.0436	10.3436	9.9322
40 - " " " " Dry Coal -	"	10.3881	10.3453	10.1626	9.7342	10.4249	10.8097	10.3756
41 - " " " " Combustible -	"	12.7178	12.6930	12.5101	12.4294	13.0042	12.9453	12.6627
42 - Dry Flue Gas Per Lb. Dry Coal -	"	15.80	13.77	14.69	13.80	14.98	16.89	15.09
EFFICIENCY								
43 - B.T.U. Per Lb. Coal as Fired, by Calorimeter -	B.t.u.	12,738	12,806	12,871	12,844	12,487	12,724	12,807
(a) B.T.U. Per Lb. Dry Coal	"	13,226	13,306	13,308	13,304	13,160	13,297	13,289
(b) " " " Combustible	"	16,190	16,147	16,161	16,178	16,116	16,168	16,162
44 - Efficiency of Unit -	%	76.00	75.45	74.14	70.48	76.62	78.89	75.71
45 - " " " Based on Combustible -	"	81.34	81.32	80.07	79.36	83.40	82.62	80.90

TABLE 1 (*Continued*)

SERIES 1002 - TEST NO.		1	2	3	4	5	6	7	8	9	10	11	12	13	1 to 13 Incl.
46 - Percent Excess Air -	%	48.08	49.04	53.47	64.06	70.35	78.08	57.21	59.69	59.63	53.80	47.90	59.63	[illegible]	[illegible]
51(a) Analysis of Dry Gases Before Damper (By Volume) -															
(1) Carbon Dioxide (CO_2)	%	12.60	11.90	11.69	10.60	10.60	10.18	11.38	11.13	12.43	11.60	11.94	11.13	10.79	11.56
(2) Oxygen (O_2)	"	6.50	7.30	7.47	8.42	8.60	9.05	7.98	8.20	6.66	7.50	7.16	8.02	8.50	7.61
(3) Carbon Monoxide (CO)	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(4) Hydrogen and Hydrocarbons	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(5) Nitrogen by Difference (N_2)	"	80.90	80.80	80.74	80.89	80.90	80.80	80.74	80.67	80.91	80.90	80.90	80.85	80.75	80.83
(b) Analysis of Dry Gases 1st Pass, (By Volume) -															
(1) Carbon Dioxide (CO_2)	%	13.40	12.50	12.23	11.44	11.60	10.97	11.90	11.50	12.65	12.50	12.57	11.72	10.80	12.30
(2) Oxygen (O_2)	"	5.50	6.50	6.70	7.54	7.48	8.23	7.21	7.54	5.91	6.50	6.46	7.36	9.40	7.05
(3) Carbon Monoxide (CO)	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(4) Nitrogen by Difference (N_2)	"	81.10	81.00	81.07	81.08	80.92	80.80	80.89	80.96	81.44	80.90	80.98	80.92	80.80	80.97
52 - Proximate Analysis of Dry Coal -															
(a) Moisture (As Fired)	%	3.74	3.90	3.67	2.68	4.00	3.88	4.07	3.72	3.76	3.24	3.68	3.66	4.81	3.90
(b) Volatile Matter	"	33.06	33.62	32.28	30.89	31.94	33.02	33.77	34.81	34.34	33.47	33.82	33.38	34.19	33.44
(c) Fixed Carbon	"	54.30	54.21	54.44	56.31	56.79	54.12	56.01	52.86	53.56	54.27	54.08	53.06	53.40	54.27
(d) Ash	"	12.14	12.17	12.88	13.00	11.27	12.06	11.22	12.93	12.10	12.26	12.10	12.67	12.38	12.29
53 - Ultimate Analysis of Dry Coal -															
(a) Carbon (C)	%	74.07	74.15	74.30	72.70	74.92	74.56	75.86	73.81	74.28	73.60	73.81	73.41	74.06	74.08
(b) Hydrogen (H)	"	5.02	5.10	5.14	5.13	5.15	5.15	5.26	5.04	5.12	5.09	5.16	5.02	5.02	5.10
(c) Oxygen (O_2)	"	6.32	6.10	5.69	5.93	6.12	5.90	5.62	6.00	6.12	6.30	5.62	6.24	6.05	6.12
(d) Nitrogen (N_2)	"	1.19	1.56	1.37	1.38	1.31	1.17	1.39	1.32	1.04	1.22	1.30	1.35	1.24	1.20
(e) Sulphur (S)	"	1.56	1.32	1.35	1.21	1.23	1.28	1.16	1.39	1.14	1.10	1.23	1.23	1.35	1.31
(f) Ash	"	12.14	12.17	12.28	13.00	11.27	12.06	11.22	12.93	12.10	12.84	12.10	12.67	12.38	12.29
54 - Analysis of Ash and Refuse, Etc. -															
(a) Combustible	%	30.33	36.51	22.94	25.61	26.15	22.96	21.16	30.30	34.60	34.67	44.36	35.11	25.26	31.75
(b) Non-combustible	"	69.67	63.49	77.06	74.39	73.85	77.04	78.84	69.70	65.40	65.33	55.64	64.89	74.74	68.25
55 - Heat Balance Based on Dry Coal -															
(a) Heat Absorbed by Boiler (Item 40 x 970.4)	%	74.08	75.33	70.53	76.44	76.90	77.74	76.76	74.00	75.43	74.14	70.63	76.02	76.09	76.71
(b) Loss due to Evaporation of Moisture in Coal	"	.37	.40	.36	.26	.39	.37	.39	.37	.37	.31	.36	.36	.42	.36
(c) " " " Burning of Hydrogen	"	4.33	4.37	4.30	4.42	4.33	4.20	4.36	4.29	4.38	4.32	4.40	4.30	4.17	4.33
(d) " " " Dry Flue Gases	"	10.86	11.16	11.65	12.14	11.87	11.63	11.21	11.71	10.76	11.24	10.80	11.33	10.87	11.33
(e) " " " Carbon Monoxide	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(f) " " " Combustible in Ash and Refuse	"	5.79	7.60	4.08	5.23	4.32	3.93	3.84	6.30	7.08	7.15	10.86	7.72	4.56	6.33
(g) " " " Heating Moisture in Air	"	.30	.31	.31	.51	.49	.45	.46	.36	.37	.37	.42	.47	.42	.40
(h) " " " Radiation and Unaccounted for -	"	3.70	.76	.75	.94	1.76	1.40	3.86	1.07	1.70	2.47	2.93	Minus 1.00	.06	1.70
TOTAL -		100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00

NOTE - Official Analyses Used.

TABLE 2 DATA AND RESULTS OF EVAPORATIVE TEST (WITH AIR PREHEATER)

SERIES 1001 - TEST NO.		4	5	6	7	8	9	10	11	4 to 11 Incl.
8 - Date Started (4:00 P.M.)		7/14/23	7/15/23	7/16/23	7/17/23	7/18/23	7/19/23	7/20/23	7/21/23	7/14 to 7/22, '23
9 - Duration (Hours)		24	24	24	24	24	24	24	24	192
10 - Kind and Size of Coal		HARWICK RUN OF MINE - BITUMINOUS								
AVERAGE PRESSURE AND TEMPERATURES										
11 - Steam Pressure Superheater Outlet, #/Sq."	Ga.	265.1	266.6	268.4	267.5	269.6	266.9	266.6	268.9	267.5
(a) Steam Pressure Drum	"	272.2	275.3	277.6	276.9	278.5	276.0	274.2	276.1	275.9
(b) Barometric Pressure	"Hg.	29.19	29.21	29.31	29.45	29.46	29.40	29.32	29.23	29.32
12 - Steam Temperature at Superheater Outlet -	°F	597.2	603.0	616.2	611.4	614.6	616.1	608.3	591.3	607.3
(a) Corresponding Saturation Temperature	"	411.0	411.5	412.1	411.8	412.5	411.6	411.5	412.2	411.8
(b) Superheat	"	186.2	191.5	204.1	199.6	202.1	204.5	196.8	179.1	195.5
13 - Feed Water Temperature at Boiler -	°F	119.1	120.4	116.7	118.4	121.7	116.6	114.9	119.7	118.4
(a) Heat in Steam above Feed Water Temperature	B.T.U./#	1223.2	1224.7	1235.0	1230.9	1229.1	1235.1	1232.9	1219.5	1229.4
14 - Average Flue Gas Temperatures -										
(a) After Preheater	°F	292	276	278	322	339	323	322	273	305
(b) Before Damper	"	459	467	473	479	508	498	489	459	482
(c) Top 3rd Pass	"	482	486	508	526	533	522	512	475	505
(d) Bottom " "	"	529	535	562	579	585	575	564	517	556
(e) Top 2nd "	"	620	659	695	698	710	696	671	594	568
(f) " 1st "	"	626	658	697	702	717	704	679	597	673
(g) After Superheater	"	885	928	986	985	1005	986	941	835	944
(h) Before "	"	1089	1130	1183	1186	1229	1208	1139	1002	1142
(i) Temperature of Front Wall by Optical Pyrometer	"	1883	2054	2019	2099	2134	2087	2053	1952	2035
(j) " " Fuel Bed " " "	"	2701	2619	2633	2716	2739				
(k) " " Bridge Wall " " "	"	2053	3089	2207	2368	2378				
(l) Average Temperature of Boiler Casing	"	139	140	144	157	162	161	168	163	154
(m) Before Damper (Foxboro)	"	500	509	528	533	538	529	518	478	517
(n) " Horiz. Circ. Tubes (Bailey)	"	474	482	495	501	506	502	485	458	488

TABLE 2 (*Continued*)

SERIES 1001 - TEST NO.		4	5	6	7	8	9	10	11	4 to 11 Inc.
15 - Drafts by Manometer -										
(a) Overfire	" Water	.06	.02	.02	.00	.00	.01	.01	.03	.02
(b) Before Superheater	"	.06	.01	.03	.02	.01	.01	.01	.01	.02
(c) After "	"	.03	.01	.02	.02	.02	.01	.01	.00	.02
(d) Top 1st Pass	"	.08	.05	.08	.08	.07	.06	.05	.00	.06
(e) " 2nd "	"	.09	.05	.10	.09	.08	.07	.06	.00	.07
(f) Bottom " "	"	.27	.25	.33	.33	.31	.29	.26	.15	.27
(g) Top 3rd "	"	.39	.40	.51	.52	.50	.46	.41	.19	.42
(h) Before Damper	"	.44	.44	.58	.60	.58	.52	.48	.23	.48
(i) " Preheater	"	.70	1.08	1.10	1.12	1.13	1.12	1.11	1.15	1.06
(j) After "	"	.68	1.05	1.09	1.10	1.11	1.10	1.08	1.09	1.04
(k) At Base of Stack	"	.82	1.11	1.23	1.27	1.29	1.26	1.26	1.21	1.18
(l) " Fan Suction	"	1.74	1.95	2.12	2.20	2.17	2.21	2.09	1.59	2.01
(m) Pressure at Fan Discharge	"	Plus 4.63	Plus 5.21	Plus 4.84	Plus 4.80	Plus 4.82	Plus 4.78	Plus 4.84	Plus 5.54	Plus 4.94
(n) " Forced Draft Duct	"	" 4.03	" 4.60	" 5.04	" 5.05	" 4.59	" 4.81	" 4.79	" 4.25	" 4.66
(o) " " Duct to Wind Box	"	" 4.65	" 5.49	" 5.20	" 5.07	" 5.08	" 5.02	" 5.19	" 5.78	" 5.19
(p) " Under Grate	"	" 3.23	" 3.90	" 4.30	" 4.55	" 4.65	" 4.48	" 3.74	" 2.20	" 3.88
(q) Bridge Wall Air Box (Pressure)	"	" .02	" .28	" .62	" .55	" .85	" .17	" .03	" .04	" .32
16 - State of Weather -		Rain	Cloudy	Clear	Clear	Clear	Clear	Clear	Clear	
(a) Temperature of External Air	°F	74	73	74	73	76	78	79	78	76
(b) " " Air at Main Gauge Board	"	80	81	82	82	84	84	89	85	83
(c) " " " " Draft " "	"	86	89	90	90	91	93	96	96	91
(d) " " " in Main Air Duct	"	105	106	104	104	106	106			105
(e) " " Inlet Air to Preheater	"	104	105	104	105	108	111	116	116	108
(f) " " Air at Fan Discharge	"	225	215	229	235	240	236	237	216	229
(g) " " Windbox	"	214	215	225	233	236	234	233	214	226
(h) " " Coal	"	85	85	81	80	75	77	82	80	81
(i) Relative Humidity	%	30.5	34.0	21.5	21.0	21.5	16.0	18.0	18.0	22.6
(j) Dry Bulb Temperature	°F	111.0	106.0	103.0	107.7	113.5	118.5	122.0	120.5	112.8
TOTAL QUANTITIES										
19 - Total Weight of Coal as Fired -	lbs.	244,340	278,385	312,820	328,155	330,965	298,675	266,580	195,410	2,255,330
(a) Average revolutions of Stoker Sections		678	781	868	937	947	850	757	554	6,372
(b) Total Ram Strokes		11,543	13,272	14,761	15,930	16,100	14,488	12,871	9,418	108,383
(c) Weight od Dry Coal per Stroke	lbs.	20.3	20.1	20.6	19.9	19.7	19.8	19.8	20.0	20.0
20 - Moisture in Coal as Fired -	%	3.99	3.94	2.89	3.61	3.96	4.16	4.37	3.65	3.81
21 - Total Weight of Dry Coal -	lbs.	234,591	267,417	303,780	316,309	317,859	286,250	254,930	188,278	2,169,414
22(a) Ash and Refuse (Weighed) -										
(1) Total Weight Wet	lbs.	42,610	47,720	48,605	49,790	48,250	39,655	41,250	33,190	351,050
(2) " " Dry	"	30,803	35,938	38,077	38,712	38,470	31,621	32,625	25,062	271,298
(3) Moisture by Analysis	%	27.71	24.69	21.66	22.25	20.27	20.22	20.91	24.52	22.72
(4) Ash and Refuse Based on Dry Coal	"	13.13	13.44	12.53	12.24	12.10	11.05	12.80	13.31	12.51
22(b) Ash and Refuse (From Analysis)										
(1) Ash and Refuse Based on Dry Coal	%	16.56	14.69	14.73	14.05	13.97	12.96	13.56	15.38	14.40
(2) Total Weight Dry	lbs.	38,848	39,284	44,747	44,441	44,405	37,098	34,543	28,957	312,323
23 - Total Combustible Burned (From 22-b)	lbs.	195,743	228,133	259,033	271,868	273,454	249,152	220,387	159,321	1,857,091
24 - Sprinkler Water to Ash Pit	Gals.		129,185	100,755	81,600	149,635	18 710	133,180	138,100	
(a) Total Revolutions of Clinker Grinder Rolls		28	27	36	39	26	26	26	27	235

TABLE 2 (*Continued*) DATA AND RESULTS OF EVAPORATIVE TEST (WITH AIR PREHEATER)

SERIES 1001 - TEST NO.		4	5	6	7	8	9	10	11	4 to 11 Incl.
25 - Total Weight of Water -										
(a) By Tank Measurement	lbs.	2,100,354	2,414,908	2,685,709	2,835,511	2,871,237	2,635,180	2,331,279	1,714,784	19,588,962
(b) " Venturi Meter Integrators	"					2,850,987	2,613,319	2,315,519	1,717,340	
(c) Difference from Tank Measurement	%					Minus .72	Minus .83	Minus .68	Plus .15	
(d) By Venturi Meter Planimetered Charts	lbs.	2,143,000	2,416,100	2,690,100	2,795,900	2,886,800	2,623,300	2,332,800	1,733,400	19,621,400
(e) Difference from Tank Measurement	%	Plus 2.03	Plus .05	Plus .16	Minus 1.40	Plus .54	Minus .45	Minus .07	Plus 1.09	Plus .17
(f) By Bailey Integrators	lbs.	2,124,000	2,447,800	2,727,700	2,868,500	2,900,000	2,651,300	2,348,200	1,715,300	19,782,500
(g) Difference from Tank Measurement	%	Plus 1.13	Plus 1.36	Plus 1.56	Plus 1.16	Plus 1.00	Plus .62	Plus .73	Plus .31	Plus .99
26 - Total Water Evaporated -	lbs.	2,100,354	2,414,908	2,685,709	2,835,511	2,871,237	2,635,180	2,331,279	1,714,784	19,588,962
27 - Factor of Evaporation -		1.2605	1.2621	1.2727	1.2684	1.2666	1.2728	1.2705	1.2567	1.2669
28 - Total Equivalent Evaporation From and At 212 °F -	lbs.	2,647,496	3,047,855	3,418,102	3,596,562	3,636,709	3,354,057	2,961,890	2,154,969	24,817,640
HOURLY QUANTITIES AND RATES										
29 - Dry Coal Per Hour -	lbs.	9,775	11,142	12,656	13,180	13,244	11,927	10,622	7,845	11,299
(a) Dry Coal Per Retort Per Hour	"	575	655	745	775	779	702	625	461	665
30 - Dry Coal Per Sq. Ft. of Grate Surface, Per Hour, -	"	44.77	51.04	57.98	60.37	60.67	54.63	48.66	35.94	51.76
31 - Water Evaporated Per Hour, -	"	87,515	100,621	111,905	118,146	119,635	109,799	97,137	71,449	102,026
32 - Equivalent Evaporation From and At 212 °F, Per Hour -	"	110,312	126,994	142,421	149,857	151,530	139,752	123,412	89,790	12[illegible],259
33 - " " Per Sq.Ft. Heating Surface " " -	"	4.81	5.54	6.22	6.54	6.61	6.10	5.39	3.92	5.64
34 - Dry Flue Gas Per Hour -	"	153,272	165,459	185,313	193,219	187,270	180,097	170,483	126,697	171,067
CAPACITY										
36 - Percent of Rating -	%	139.6	160.7	180.2	189.6	191.7	176.8	155.1	115.6	163.5
ECONOMY										
37 - Water Evaporated Per Lb. Coal as Fired -	lbs.	8.5960	8.6747	8.5855	8.6408	8.6753	8.8229	8.3700	8.7752	8.6856
38 - " " " " Dry Coal -	"	8.9535	9.0305	8.8410	8.9644	9.0331	9.2059	9.1446	9.1077	9.0296
39 - Equivalent Evaporation Per Lb. Coal as Fired -	"	10.8353	10.9483	10.9267	10.9599	10.9882	11.2298	11.1107	11.0279	11.0040
40 - " " " " Dry Coal -	"	11.2856	11.3974	11.2519	11.3704	11.4413	11.7172	11.6184	11.4457	11.4398
41 - " " " " Combustible -	"	13.3524	13.3560	13.1956	13.2291	13.2992	13.4619	13.4395	13.5260	13.3637
42 - Dry Flue Gas Per Lb. Dry Coal -	"	15.68	14.85	14.64	14.66	14.14	15.10	16.05	16.15	15.14
EFFICIENCY										
43 - B.T.U. Per Lb. Coal as Fired, by Calorimeter -	b.t.u.	12,659	12,884	12,929	13,086	12,879	12,953	12,850	12,631	12,895
(a) B.T.U. Per Lb. Dry Coal,	"	13,185	13,412	13,314	13,576	13,410	13,515	13,437	13,317	13,406
(b) B.T.U. " " Combustible,	"	15,172	15,186	15,191	15,260	15,193	15,207	15,209	15,165	15,201
44 - Efficiency of Unit -	%	85.06	82.46	82.01	81.27	82.79	84.13	83.91	83.40	82.81
45 - Efficiency of Unit Based on Combustible -	"	86.51	85.35	84.29	84.13	84.94	85.90	85.75	86.55	85.31

TABLE 2 (*Continued*)

SERIES 1001 - TEST NO.		4	5	6	7	8	9	10	11	4 to 11 Incl.
46 - Excess Air -	%	59.27	49.21	46.24	45.06	40.35	47.54	57.23	60.77	50.49
51(a) Analysis of Dry Gases Before Damper (By Volume) -										
(1) Carbon Dioxide (CO_2)	%	11.24	12.03	12.27	12.34	12.90	12.40	11.40	11.10	11.96
(2) Oxygen (O_2)	"	8.15	7.32	6.94	6.84	6.10	6.80	7.86	8.20	7.28
(3) Carbon Monoxide (CO)	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(4) Hydrogen and Hydrocarbons	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(5) Nitrogen by Difference (N_2)	"	80.61	80.65	80.79	80.82	81.00	80.80	80.74	80.70	80.76
(b) Analysis of Dry Gases 1st Pass (By Volume) -										
(1) Carbon Dioxide (CO_2)	%	11.47	12.18	12.40	12.92	13.03	12.39	11.90	10.88	12.15
(2) Oxygen (O)	"	7.70	6.89	6.70	6.10	6.05	6.74	7.28	8.42	6.98
(3) Carbon Monoxide (CO)	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(4) Nitrogen by Difference (N_2)	"	80.83	80.93	80.90	80.98	80.92	80.87	80.82	80.70	80.87
52 - Proximate Analysis of Dry Coal -										
(a) Moisture (As Fired)	%	3.99	3.94	2.89	3.61	3.96	4.16	4.37	3.65	3.81
(b) Volatile Matter	"	34.31	34.95	33.72	33.73	33.74	32.81	32.06	34.17	33.66
(c) Fixed Carbon	"	52.61	53.37	53.93	55.09	54.64	56.06	56.29	53.64	54.53
(d) Ash	"	13.08	11.68	12.35	11.18	11.62	11.13	11.65	12.19	11.81
53 - Ultimate Analysis of Dry Coal -										
(a) Carbon (C)	%	73.69	73.97	73.67	74.72	74.58	76.10	74.75	74.67	74.53
(b) Hydrogen (H)	"	4.95	5.15	5.09	5.12	4.86	4.61	5.16	5.11	5.00
(c) Oxygen (O)	"	5.81	6.56	6.36	6.59	6.48	5.76	6.08	5.48	6.19
(d) Nitrogen (N)	"	1.24	1.32	1.40	1.30	1.27	1.15	1.24	1.37	1.28
(e) Sulphur (S)	"	1.23	1.32	1.13	1.09	1.19	1.25	1.12	1.18	1.19
(f) Ash	"	13.08	11.68	12.35	11.18	11.62	11.13	11.65	12.19	11.81
54 - Analysis of Ash and Refuse, Etc. -										
(a) Combustible	%	21.02	20.51	16.18	20.44	16.82	14.09	14.01	20.73	17.96
(b) Non-combustible	"	78.98	79.49	83.82	79.56	83.18	85.91	85.99	79.27	82.04
55 - Heat Balance Based on Dry Coal -										
(a) Heat Absorbed by Boiler	%	83.06	82.46	82.01	81.27	82.79	84.13	83.91	83.40	82.81
(b) Loss due to Evaporation of Moisture in Coal	"	.38	.37	.27	.34	.38	.40	.42	.35	.36
(c) " " " Burning of Hydrogen	"	4.11	4.22	4.22	4.21	4.08	3.81	4.26	4.22	4.15
(d) " " " Dry Flue Gases	"	7.00	6.69	6.56	6.90	6.88	7.08	7.34	7.13	6.[illegible]
(e) " " " Carbon Monoxide	"	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
(f) " " " Combustible in Ash and Refuse	"	3.85	3.28	2.61	3.09	2.56	1.98	2.06	3.50	2.82
(g) " " " Heating Moisture in Air	"	.23	.22	.12	.14	.17	.15	.19	.17	.18
(h) " " " Radiation and Unaccounted for	"	1.37	2.76	4.21	4.05	3.14	2.45	1.82	1.23	2.[illegible]
TOTAL -		100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00

NOTE - Official Analyses Used.

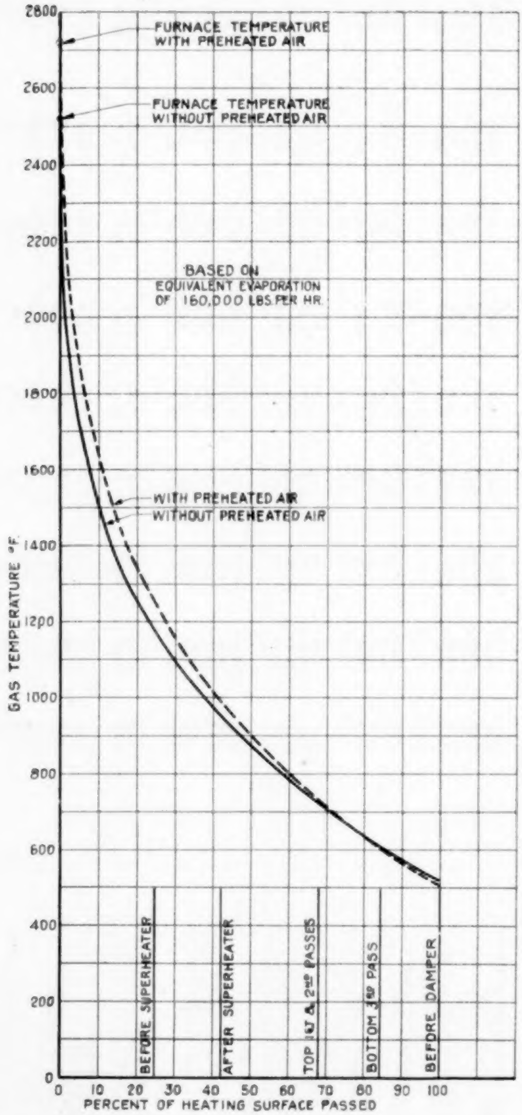

Fig. 12 Comparative Temperature Drop through Boilers

is approximately 15 deg. lower on the average than the same reading in the data without air preheater. This is due to the higher furnace temperatures and better CO_2, resulting in a more efficient heat transfer in the boiler. It will also be observed that Item 14 (*b*) does not check with Item 14 (*m*) due to the fact that Item 14 (*m*) is a reading from the permanent recording-thermometer installation with the element placed in the center of the pass, whereas Item 14 (*b*) is an average of six readings across the path. Readings 14 (*e*) to 14 (*i*), inclusive, are, of course, subject to the radiation effect of the cold tubes, but they are comparative for the runs with and without the preheaters. The temperature reading below the horizontal circulating tubes, Item 14 (*n*), by recorder is

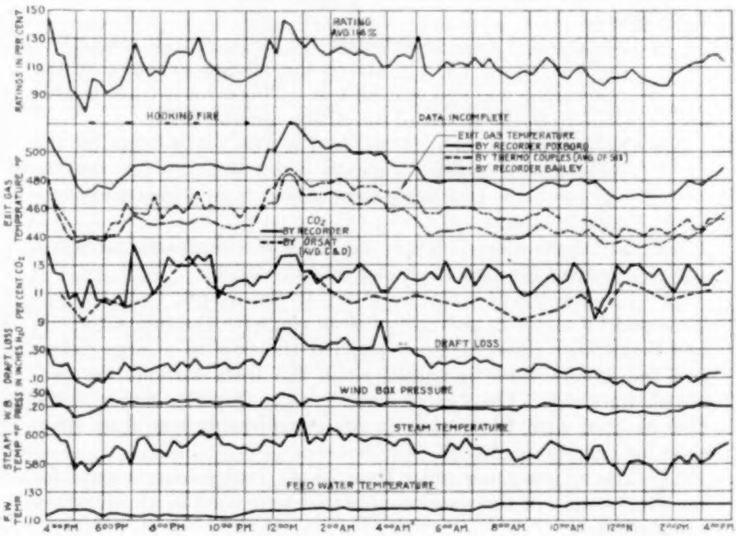

FIG. 13 LOG OF TEST NO. 11 WITH PREHEATED AIR

lower than the reading before the damper due to the heat absorption through the tubes.

27 Draft readings before the preheater, Item 15 (*i*), are somewhat inaccurate on account of the eddy currents set up by the boiler damper, which tend to stratify the gases at this point. The draft readings over the preheater, Item 15 (*j*), are very low due to the air infiltration referred to previously.

28 It will be noted that the pressure readings at the fan discharge, Item 15 (*m*), and the pressures in the duct to the wind box, Item 15 (*o*), do not correspond. This is due to the fact that the duct to the wind box has a considerably larger cross-section than the fan discharge, resulting in a decrease in dynamic head and a higher pressure indication.

29 The temperature of the preheated air, Item 16 (*f*), was taken at the fan discharge rather than at the preheater discharge because the fan decreased the possible error due to stratification. As the preheated-air duct between the preheater and the fan suction is effectively insulated, the drop in temperature between these two points was considered negligible.

30 The weight of dry coal per stroke of the stoker rams, Item 19 (*c*), remained fairly constant, thus forming a check on the general accuracy of coal measurement.

31 The weight of ashpit refuse is calculated from actual weights under Item 22 (*a*) and from coal and ash analyses under Item 22 (*b*). However, the ashpit losses are figured entirely from the laboratory analyses. The weighed quantities are not satisfactory for at least three reasons:

a It is possible only to estimate the difference in amount of refuse left in the pit above the clinker grinders at the beginning and end of each test. This error is considerable due to the large size of the clinker-grinder pit

b Considerable error is introduced in the ash weights by the presence of moisture, which varies from 20 per cent to 35 per cent, this being determined from a comparatively small sample

c No definite determination was made of the amount of refuse carried through the boiler into the stack.

32 The amount of sprinkler water to the ashpit, Item 24, was measured by meter in the case of tests run with the preheater. It was found that these meters restricted the flow to a large extent and the meters were by-passed when the tests without the preheater were run. The figures given for the sprinkler water in the case of tests with the preheater are considerably less than those used for general operation.

33 The figures given under Item 25 (*a*), tank measurements, for the total weight of the water evaporated were used in all efficiency calculations.

34 The weight of dry flue gas per hour, Item 34, was calculated from the CO_2 readings. These CO_2 readings, although affording a fair indication of the relative conditions for the various tests, cannot be considered as strictly accurate due to the variation in gas composition across the boiler and from front to back in the pass. This variation in gas composition across a section of one of the passes varies as much as 2 per cent in actual value of CO_2 and from front to back of the pass as much as 4 per cent in actual value of CO_2. With an average CO_2 value of 12 per cent these figures correspond to a variation in excess air of 22 per cent and 50 per cent, respectively, so that it is extremely difficult if not impossible to arrive at results actually representative of average conditions across the pass.

35 The analyses of dry gases before the damper, Item 51 (*a*), show a complete absence of CO, H, and hydrocarbons. The anal-

ysis of gases in the center of the first pass, Item 51 (*b*), shows no indication of CO. The fact that no carbon monoxide was found in the first pass indicates that combustion is complete shortly after the gas enters the tubes.

36 Figures shown under Items 43, 52, and 53 are the official analyses reported by the Duquesne Light Company laboratories, checked by independent analyses made in the laboratories of the Babcock & Wilcox Company. The determinations shown are approved by the latter company.

37 Tests Nos. 6 to 10, inclusive, as shown in the data and results

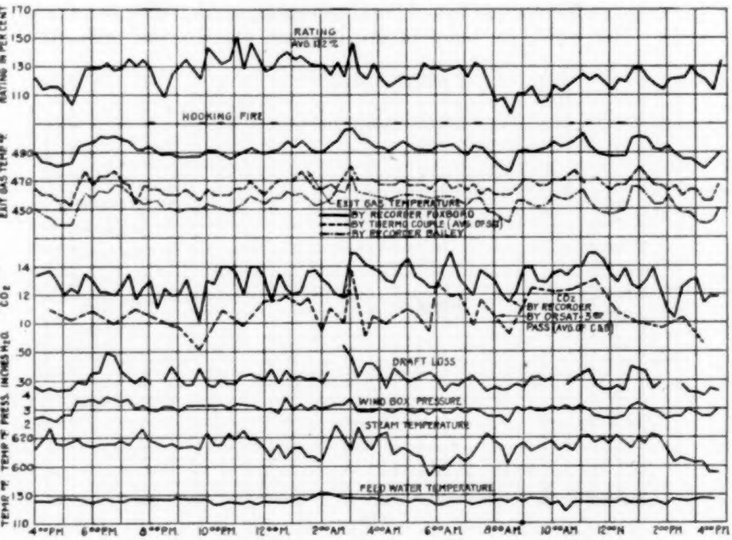

Fig. 14 Log of Test No. 13 without Preheated Air

for Boiler No. 9, without preheater, were run using the preheater fan for forced draft. The air inlets to the preheater were blocked off and doors in the suction duct to the fan were opened to take air directly from the boiler room. This was done on account of the closer air control possible by the use of the single fan.

38 Tests Nos. 10, 11 and 12 were intended for high-rating runs, but the ratings planned were not reached due to excessive clinkering. Operation above 220 per cent of rating is not at all reliable under the present conditions.

39 The reduction of gas temperature through the preheater in the runs without the preheater operating, as indicated under Items 14 (*a*) and (*b*), is due to air leakage and radiation, there being no air drawn through the preheater by the fan during these tests. Readings 15 (*l*) and 15 (*m*) are not complete, owing to the fact that

tests Nos. 1 to 5 and Nos. 11 to 13, inclusive, were run on the main air duct.

40 Curves shown in Fig. 9 indicate comparative boiler performance with and without preheated air. The increase in efficiency due to preheated air as shown by the curves varies from $5\frac{1}{2}$ per cent at 114 per cent rating to 7 per cent at 200 per cent rating, the general effect being a tendency to flatten out the curve in the case of the preheated-air tests. This is due to the increase in heat transmission of the preheater with the increase in temperature and volume of flue gas and air through the preheater, and also to the marked decrease in the percentage of combustible in the ash as indicated in the comparison of curves showing combustible in refuse.

41 The wind-box temperature increase due to the preheater

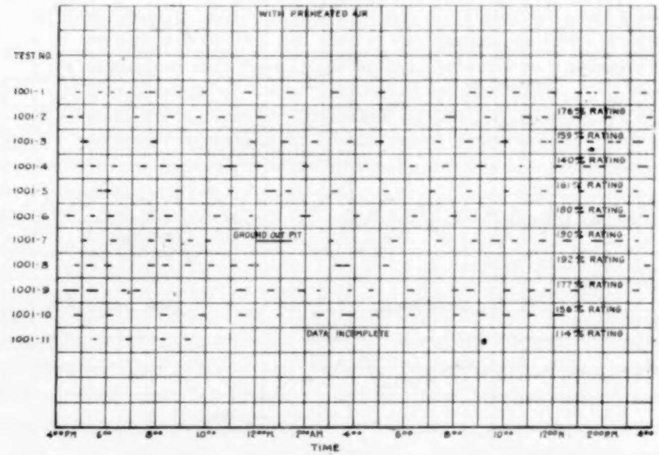

FIG. 15 LOG OF HOOKING FIRES WITH PREHEATED AIR

varies from 120 deg. fahr. for 114 per cent rating to 140 deg. fahr. for 200 per cent rating. This higher temperature is the limit that can be reached with the preheater equipment as installed.

42 The exit gas temperatures are lowered in the case of tests with preheated air, the amount of reduction varying from 15 deg. fahr. at the lowest rating to 10 deg. fahr. at 200 per cent of rating. The reduction is explained by the higher mean gas temperature through the boiler, resulting in more effective heat transfer. This increase in mean gas temperature is due to the decrease in gas quantity as indicated by the increased CO_2 and the higher furnace temperatures using preheated air.

43 The curves indicate an average increase of CO_2 for the results with preheated air varying from 0.7 per cent at the low rating to 1.4 per cent at 200 per cent rating. This CO_2 increase is due to better combustion conditions. The fuel bed was much more

uniform and there was an almost entire absence of air holes or disturbances of any kind to break up the uniformity of stoker operation.

44 It will be noted that the percentage of combustible in the ash remains practically constant for all ratings when preheated air is used, the decrease of percentage of combustible over that for operation without preheated air varying from 6 per cent to 12 per cent.

45 No material change in superheat is indicated.

46 The increase in fuel-bed and front-wall temperatures for the test with preheated air over that without, shown by curves in Fig. 10, is considerably greater than the increase of the wind-box air tem-

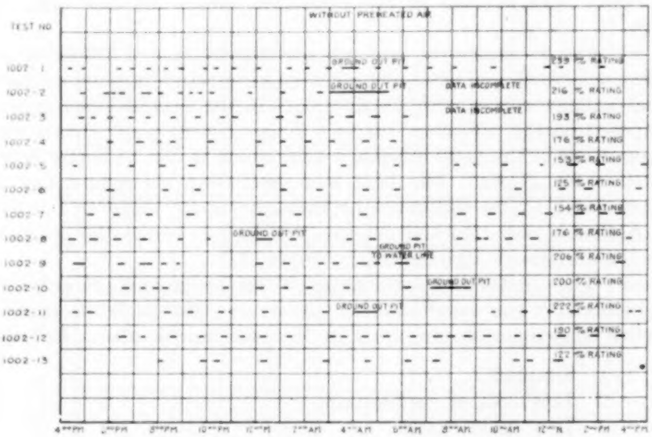

Fig. 16 Log of Hooking Fires without Preheated Air

perature on account of the reduction in excess air. The remaining curves in Fig. 10 are interesting in that they indicate the relative gas temperatures throughout the boiler.

47 The curves in Fig. 11 show a decrease in draft loss through the boiler for preheated-air operation, varying from about 20 per cent reduction for the low rating to 30 per cent for 200 per cent rating.

48 Fig. 12 indicates the temperature drop through the boiler with and without preheated air for conditions which obtain at 200 per cent of rating. The curves clearly show that the mean temperature for the total travel of gases is higher with preheated air, even though the final exit temperature is lower.

49 Figs. 13 and 14 show curves of a typical test log with and without preheated air, respectively. These curves show the variation of the load on the boiler during test, with corresponding

variations in the other principal readings, and are a representative indication of the continuity of the records.

50 The charts shown in Figs. 15 and 16 indicate the frequency of hooking fires for tests with and without preheated air, respectively. These charts show that grinding out of the ashpit was less frequently required in the case of tests using preheated air.

51 Curves in Fig. 17 show special CO_2 traverses at the top of the third pass and in the uptake before and after the preheater. As noted previously, such traverses were taken once for each test. The curves represent a typical example of the conditions existing. The decrease in CO_2 between traverse in the third pass and traverse in the uptake before the preheater is due to air leakage at the expansion joint between the boiler damper frame and the uptake. The leakage in the preheater causes another drop in CO_2.

52 Fig. 18 shows the approximate distribution of difference in efficiencies with and without preheated air plotted from cumulative heat-balance curves with the difference in the various losses superimposed on the efficiency curves. These curves show clearly the proportion of the total efficiency gain which is due to better combustion conditions. The heat recovered from the ashpit loss is represented by the difference between curves *EF* and *CD*, but the variation in radiation and unaccounted-for loss brings the net gain to the efficiencies represented by curve *AB*.

53 The outstanding advantages of the flue-gas air preheater as evidenced by the installation at Colfax, in addition to the gain in boiler efficiencies as presented in the test results, are as follows:

1 Simplicity
2 Low cost of installation
3 Low maintenance
4 Small operating charges.

As previously pointed out, the preheater equipment can often be installed without any additional building space. The comparatively low cost of the equipment itself is evident from the fact that the total weight of material required to produce the same effect with water economizers is considerably greater and requires the use of an induced-draft fan for the flue gas and a special pumping system for the feedwater.

54 As to maintenance, the preheater installation at Colfax perhaps has not been operated long enough to establish definitely reliable data. However, the only item questionable is the preheater-element construction. There has been, up to the present time, absolutely no indication of deterioration in this connection. Furthermore, the reduced amount of equipment required for the preheater installation as compared to an economizer installation reduces maintenance charges.

55 The only operating cost chargeable to the preheater installation is the power required for moving air through the preheater

and duct system. In the Colfax installation this amounts to about 5 kw. per ton of coal fired, or the equivalent of less than 0.4 per cent of the coal used in the boiler.

56 In addition to this, the more even furnace conditions, resulting in a less troublesome and smoother operation, constitute

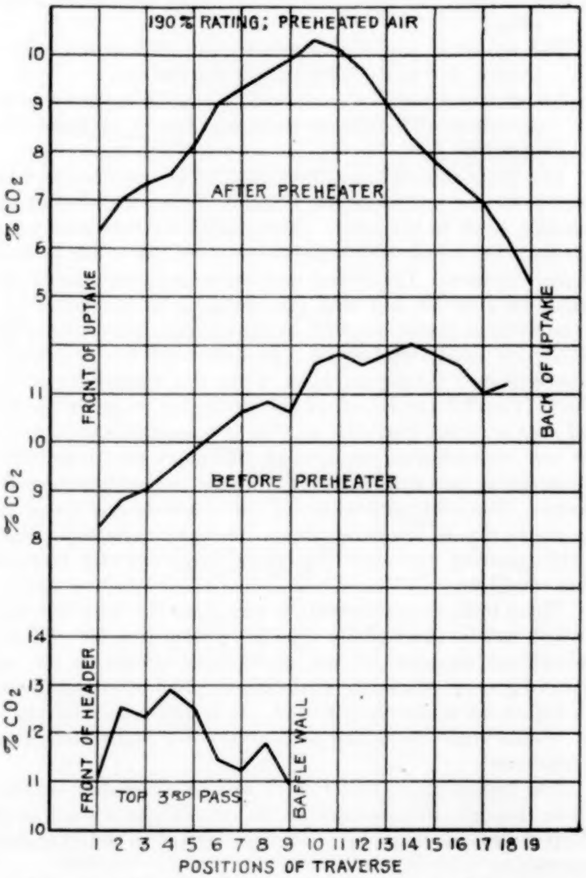

FIG. 17 CO_2 TRAVERSES BEFORE AND AFTER PREHEATER

an asset that is quite real even though it may be difficult to capitalize its value. In this connection, the notes made by the Westinghouse stoker engineers, covering conditions observed during these tests, include the following items:

a The fuel bed was much more uniform with preheated air and no holes or disturbances of any kind appeared to break up the uniformity of operation

b The fuel ignited more readily

c The fuel bed burned much more uniformly throughout its depth

d There was less combustible ejected from the underfeed section to the clinker-grinder pit

e Less carbon was contained in the refuse discharged into the ashpit

f The refuse in the clinker-grinder pit ground out with less trouble due to the character of the clinker

g No clinker trouble of a serious nature was encountered at any time. The clinker itself was harder and more easily handled.

57 The preheater equipment was placed in operation on May 7, and was in service until June 13, when the boiler was shut down for overhauling prior to the tests. Some readjustments were made in the spacing of elements which had been forced out of line by uneven expansion stresses. The boiler and preheater were placed on the line again on July 10, and with the exception of the period for the tests run without preheater, were in continuous operation supplying preheated air until October 19. Trouble with the preheater fan then necessitated taking air again from the regular forced-draft-fan duct. Careful inspection of the preheater revealed no indication of soot or other deposits, so that it was unnecessary to make use of any soot-blowing equipment, although the installation of such equipment was considered at the time the equipment was put in service. The vertical position of the elements and the weaving of the plates due to temperature and pressure variations, together with the scouring action of the gases, are evidently responsible for this condition.

58 These tests do not indicate in any sense the limit in temperature which can be successfully used for combustion air. There are no indications whatever of any undesirable effects on the stoker parts, or of general combustion conditions which were not favorable to the higher air temperatures used. It is quite probable that still better results could be gained with somewhat higher temperature than that used.

59 The reduction in boiler draft loss in a measure offsets the draft loss through the preheater, so that the height of stack to offset the drop in flue-gas temperature is not so great as would ordinarily be expected.

60 Feedwater temperatures throughout the tests were low. These temperatures do not represent the normal conditions at the Colfax station, but were necessitated by the arrangement of water-measuring apparatus, which did not permit the water to pass through the heater system.

61 In working up the tests the fact was brought out that the Boiler Test Code as it stands at present is somewhat indefinite as to what normal combustion-air temperatures entering the boiler should be. The inference is that boiler-room temperature should be used for

this air when calculating efficiencies and heat balance. On account of the variation in this temperature at different positions around the boiler this figure is very indefinite. The heat balances given in the present paper were therefore figured on the basis of the actual

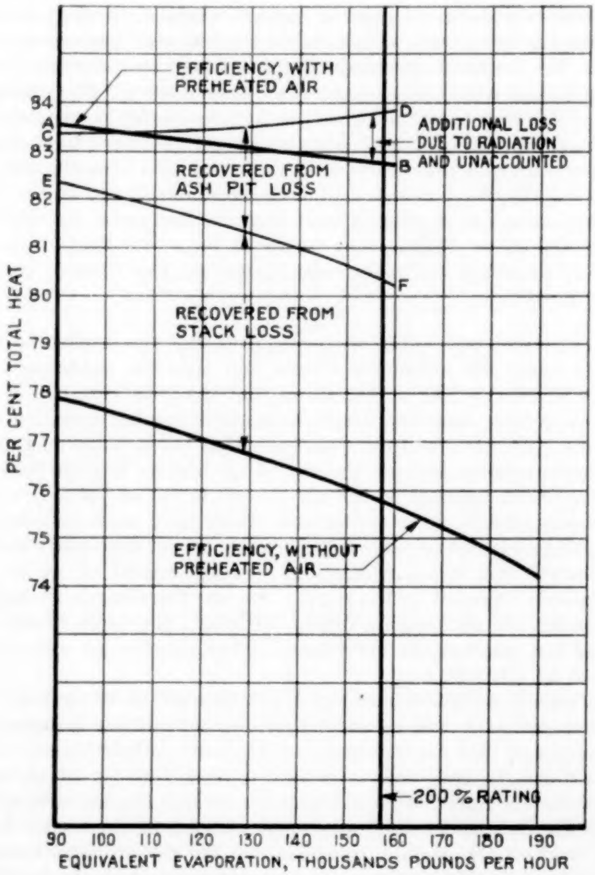

Fig. 18 Distribution of Difference in Efficiencies

air temperatures in the wind box. It is believed that a revision of the Code to define more clearly standard conditions of air temperatures should be made. This matter is especially important in comparing results with preheated air.

DISCUSSION

John H. Lawrence. It would be of interest to learn whether the boiler which was tested was in service long enough to determine whether there would be excessive maintenance on the brickwork. The writer understands that in Europe, where air preheaters have been used extensively, considerable trouble with furnace maintenance has occurred due to the higher furnace temperatures. The air preheater will be used extensively in the future. The regenerative cycle seems to be the logical one for present-day power stations. To make full use of it the feedwater should be heated to as high a temperature as is practicable with the steam bled from the turbine. This reduces the economic field of the economizer, and unless there is some means of recovering heat from the flue gases, considerable waste will ensue. There thus seems to be a wide field for the air heater, provided the maintenance cost on the furnace can be kept down.

C. F. Hirshfeld. The writer believes that the results recorded in the paper are attainable results, but that the paper does not prove this fact. The author states that the tests were run by the operating force under as nearly as possible regular operating conditions. The results of the tests show that the possible magnitude for improvement without the use of air heaters is such that the improvement credited to the air heaters is not at all a proof of their possibilities. The writer and others have made calculations on a large number of tests of air heaters, which were fairly closely performed and these calculations indicate results of about the magnitude recorded in the paper. We are therefore fairly safe in accepting the conclusions drawn, although the tests themselves would not warrant the acceptance of the conclusions unless supported by other data.

In regard to maintenance of the brickwork, it is obvious that greater difficulty will be experienced as the furnace temperature is raised and that the maintenance, therefore, will be higher. This is particularly true with stokers requiring the use of an arch. However, the refractory manufacturers are now beginning to appreciate the fact that refractories must be sold on a scientific basis. The work of the Mellon Institute and the Government bureaus, together with that of the large operating companies, is providing a fund of information which will permit the purchase of refractories on rational lines. Therefore, when air heaters come into extensive use, the refractory manufacturers probably will be able to produce satisfactory refractories and the information will be available to enable these to be chosen wisely, with the result that maintenance costs will not be prohibitive.

A. A. Adler. One danger that will confront the users of air preheaters who burn coal on ordinary stokers or hand-fired grates,

is that for nearly every coal the ash is dangerously close to the fusing point at usual rates of driving boilers. The increased temperature of the fuel bed with preheated air is likely to fuse the ash in the coal and thus diminish the air supply and cut down the boiler capacity due to the formation of clinker. The evident field for preheaters is in powdered-coal furnaces and for oil burning. Preheaters applied to ordinary plants burning coal may lead to more trouble than the increased efficiency will warrant. If by preheating the air the theoretical efficiency is improved, say, from 3 to 4 per cent, but if at the same time the coal consumption is increased by

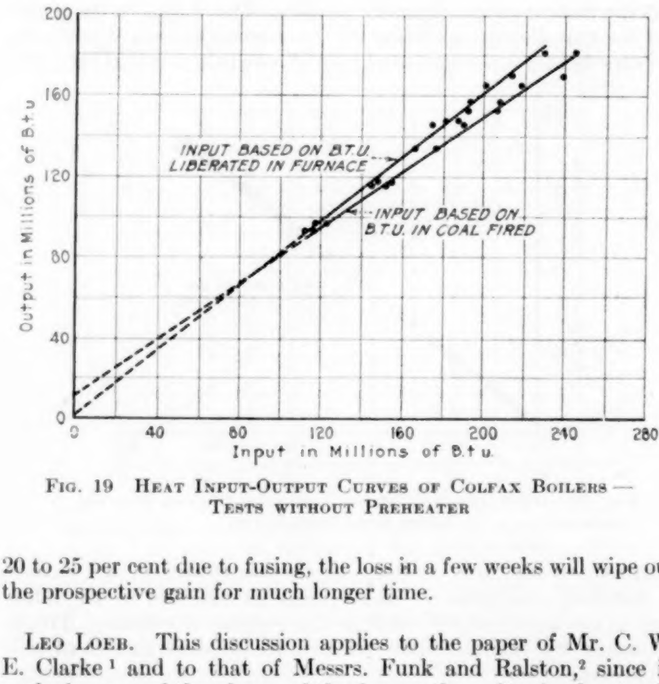

FIG. 19 HEAT INPUT-OUTPUT CURVES OF COLFAX BOILERS — TESTS WITHOUT PREHEATER

20 to 25 per cent due to fusing, the loss in a few weeks will wipe out the prospective gain for much longer time.

LEO LOEB. This discussion applies to the paper of Mr. C. W. E. Clarke [1] and to that of Messrs. Funk and Ralston,[2] since in analyzing one of the phases of the former the writer inadvertently came upon some phases of the latter.

After reading Mr. Clarke's paper this question arose: Apart from the thermal considerations, what are the purely physical features which explain the higher efficiency attained with air preheaters? In order to answer this query, the writer plotted for both series of tests a heat input-output chart, shown in Figs. 19 and 20, in which the abscissas represent the B.t.u. input per hour and the ordinates represent to the same scale the B.t.u. absorbed by the boiler and superheater per hour.

[1] Paper No. 1911, page, 567. [2] Paper No. 1912, page 607.

The two curves in each figure represent respectively the input based on heat in coal as fired and heat liberated in the furnace, that is, heat in combustible. The output at any point of the lower curve represents the combined furnace and boiler efficiency times the input, while in the upper curve the furnace efficiency has been largely if not entirely eliminated.

Fig. 21 reproduces the upper curve of the first two figures, and while there can be some slight difference of opinion as to the slopes of the lines, it will be generally admitted that the output is a linear function of the input for the operating range covered by the tests, namely, from 100 to 200 per cent of builder's rating, and that the lines are approximately parallel. This means to the writer that the heat-absorbing efficiency of the heating surface of the boiler and superheater is not increased by preheated air, but that the gains

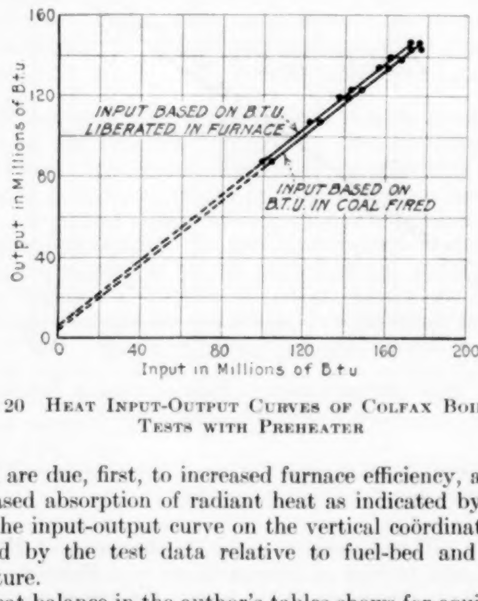

FIG. 20 HEAT INPUT-OUTPUT CURVES OF COLFAX BOILERS — TESTS WITH PREHEATER

recorded are due, first, to increased furnace efficiency, and second, to increased absorption of radiant heat as indicated by the intercept of the input-output curve on the vertical coördinate. This is confirmed by the test data relative to fuel-bed and front-wall temperature.

The heat balance in the author's tables shows for equivalent ratings an average increase in furnace efficiency (as measured by decreased loss due to combustible in ash and refuse) of 2.1 per cent, and the input-output curves show an increase in radiant-heat absorption of about four million B.t.u. per hour. In each case the heating surface efficiency is almost exactly 80 per cent.

With these results in mind, as well as many similar linear functions plotted for other makes of boilers over a wide range of ratings, it is difficult for the writer to understand how Messrs. Funk and Ralston obtained the form of input-output curve shown in Fig. 1 of their paper. It is, of course, a fact that at no load the curve has

an intercept on the input axis and must therefore bend toward that axis at low rating, but this intercept represents conditions obtaining when the apparatus is no longer a generator of steam. It follows that over the normal working range of a standard type of boiler with clean surfaces, the equation of the input-output curve is

$$Y = AX + B$$

where A is a constant depending upon the furnace efficiency and upon the heating-surface efficiency of a particular installation, and B is a function of the absorption of radiant heat and of other factors.

Possibly there is another term of higher order in the equation

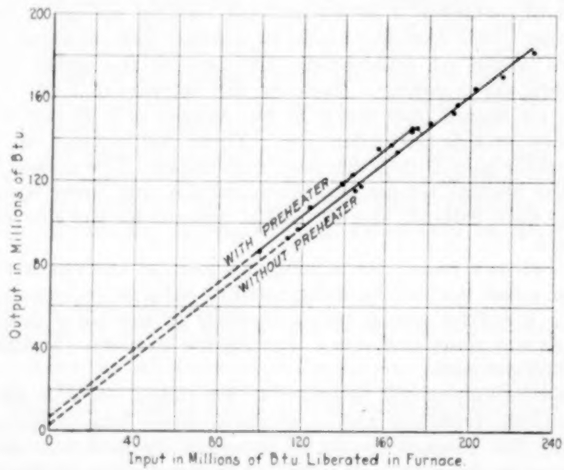

Fig. 21 Heat Input-Output Curves of Colfax Boilers Showing Effect of Air Preheaters

when an economizer is a part of the equipment. However, the writer is unable to agree with the conclusion of Par. 10.

Mr. Funk's equation does contain a final term which, taken with the first, might produce a correct result, but this has been dismissed in Par. 14 with the statement that "it is made up principally of blow-down and soot-blower losses, and these are not originally included in boiler-test results."

The writer would appreciate a fuller exposition of the reason why the coefficient of X is unity, instead of a decimal less than unity when the units are the same on both axes.

A. G. Christie. Mr. Clarke's paper presents the first recent American data on the effect of air preheaters. Some significant facts may be deduced from these tests. The tabulated data show that without preheaters the average air temperature entering the furnaces was 96 deg. fahr., while with preheaters this average was

226 deg., a difference of 130 deg. In order to determine the influence of this preheating on the boiler efficiency an estimate was made of the actual air used. This figured out about 15.15 lb. per pound of coal with preheated air. In order to reduce the two sets of tests to a comparable basis it was assumed that the flue-gas loss with preheaters could be reduced an amount equivalent to preheating the air 130 deg. fahr., which by calculation amounts to 3.6 per cent. The curves in Fig. 18 show a saving of about 4.5 per cent as recovered from the stack losses. Where did the 0.9 per cent come from?

A calculation of excess air for loads up to 200 per cent rating showed that it averaged 58 per cent without preheater and 51 per cent with preheater, a saving of 7 per cent by the use of the air preheater. This decreased excess air together with the lower carbon content of the ash undoubtedly accounts for the higher average CO_2 with the preheater. However, the decrease of 7 per cent in excess air should only result in an increase of 0.25 per cent in efficiency in this particular case. There still remains an unaccounted-for gain of 0.65 per cent in efficiency. The author states that the flue-gas temperature averaged 12½ deg. lower with preheaters than without, and this would account for the gain of 0.65 per cent.

The writer's reason for making this form of analysis is to emphasize a fact that has been suggested by other tests, namely, that there is a distinct gain in boiler efficiency by the use of preheated air over and above that due to heating the air alone. In this case it is 0.65 per cent.

The efficiency of the preheater is not stated. If it is assumed that an ideal preheater could heat the air to the temperature of the entering flue gases, then the efficiency of the preheater may be expressed as the actual air temperature rise divided by the difference between entering flue-gas and entering-air temperatures. In this case the average rise in temperature was 121 deg. fahr., and the maximum theoretical rise 374 deg. fahr., giving an efficiency of 32.4 per cent. This seems about the average efficiency to be obtained by this type of air preheater according to other data that are available to the writer.

What is needed today is a simple, cheap form of air preheater that will be permanent and will remain clean, and that will give efficiencies of the same order as the boiler efficiency, i.e., 70 to 80 per cent. The Ljungström preheater recently offered in this country has an efficiency according to recent statements in *Beama* of 70 to 75 per cent. It is, however, subject to some criticism from a mechanical point of view and it is being offered at a prohibitive price. The idea of using sheet-steel plates for regenerators is an old American idea, having been used in the Rider-Ericcson hot-air engine years ago. Some bright genius should be able to develop a regenerative form of air preheater, cheap to build, mechanically correct, and reliable in operation. Such a preheater combined with

high-stage turbine bleeding would produce a very attractive heat balance.

R. Sanford Riley. We are greatly indebted to the author for the data and analysis of the results of operating with preheated air. The increase in efficiency of $5\frac{1}{2}$ to 7 per cent is gratifying and opens up great possibilities for improved operation with stoker-fired plants.

The most impressive feature to the writer is that there are apparently no bad results in the fuel bed. There is an actual decrease of combustible in the ash, and apparently the stokers are as easy to operate with preheated air. However, it will not do to draw too positive conclusions from this particular performance. The fusing temperature of the ash in the coal is not mentioned, but we must assume that it was relatively high. What would happen with coal containing ash of low fusing temperature may be another story.

For instance, the series of evaporation tests presented before the Industrial Heating Congress in Paris in June, 1923, show that although the preheating was very moderate, only 72 deg. fahr., there was serious trouble in the fuel bed. A translation of part of a paper presented by Mr. Kammerer, Chief Engineer of the Alsacian Association of Steam-Plant Proprietors of Mulhouse, follows:

What constitutes the unexpected result is that the output obtained with air preheating has been on the average a little lower than that without air preheating. One would have had a right to expect the contrary, for, even if we agree that a rise of temperature of 72 deg. fahr. of the combustion air ought not to have a very noticeable influence on the progress of combustion in the stoker, the additional recovery of the sensible heat contained in the gases which the air preheater enables us to obtain when it is placed back of the economizer, ought to constitute a net calorific gain.

Examination of the heat balance enables us to assign a reason for this established paradox. In reality, the additional saving of heat realized by the preheater, which averages 2.5 per cent, is more than offset by the increase in losses due to combustible in the ash, which averages 3.65 per cent. Although these ought normally to be more completely burned with warm air than with cold air, we have discovered, on the contrary, how difficult it was for the fireman to avoid, during the tests of warm air, a carrying away of coke in the ashes, the latter being in a plastic state, and surrounding the pieces of coke completely. All efforts to arrive at a better combustion of the ashes have been in vain. Now, this difficulty disappeared as if by magic when we stopped preheating the air. The ashes melted less and could be burnt up perfectly at the lower end of the firebox, without the fireman having to give special attention to the matter.

We may conclude that, at the time of the run with warm air, the temperature of combustion must have reached that of fusion for the ash, just as it remained less with the feeding of cold air. The fusion temperature was in fact particularly low in the case of the fuel used; it was found in our laboratory, for the specimen of coal selected during the test, equal to 2048 deg. fahr.

This example, which should by no means be interpreted as opposing the use of warm air, shows how far, in modern installations aiming for high temperatures in the furnace, those factors which at first glance appear wholly secondary can influence the output or the conditions of the run.

From the above it will be seen that there are two sides to this preheating question, and we need more data in this country before general conclusions may be drawn. We must concede that more work has been done along this line in Europe than in this country, yet the preheating of air has not been generally adopted or conceded to be advantageous over there. It would be unfortunate if the results described by Mr. Clarke were accepted as a criterion of what may be expected of preheated air with any kind of coal. The above translation of results obtained abroad speaks for itself and should sound a note of caution and stimulate experiments in this country with other kinds of coal. We particularly need information including the temperature of fusing and other characteristics of the ash.

F. P. Fairchild.[1] Professor Christie's observations as to the causes resulting in the increase in efficiency with the air preheater are interesting. The curves shown in Fig. 18 could, of course, be further analyzed, but it was thought unnecessary to do so in this paper. He goes into the matter of efficiency of the air preheater alone and gives some theoretical figures. We did not attempt to work out the efficiency of the preheater, as there was a great deal of air infiltration which we were unable to stop. Likewise we made no attempt to determine the heat transmission of it.

Regarding Mr. Riley's discussion, it is perfectly true that the kind of coal, the fusing point of the ash, the type of stoker used, the dimensions of the furnace, the amount of the boiler surface subject to radiant heat, all affect the part of the increase in efficiency due to better combustion conditions. As stated in this paper, we are here presenting the results of tests on an actual boiler installation and have made no attempt to correct them to apply to various conditions obtaining elsewhere. The use of a relatively moderate amount of preheat on this boiler at Colfax was due directly to the fact that Mr. Clarke's investigations in Europe indicated that difficulties such as Mr. Riley cites must be considered.

Regarding Mr. Lawrence's question, I would say that this preheater installation operated continuously for about five months and we were unable to detect any tendency to speed up the deterioration of the brickwork. It is possible that some of the castings in a clinker-grinding pit might require a little more maintenance.

I am not quite sure whether Mr. Hirshfeld intends to question whether these tests were results which can actually be obtained day in and day out. Our daily checks on the operation of the boilers bear out approximately the results that this paper discussed.

Regarding Mr. Hirshfeld's further remarks, the same answer applies as in the case of Mr. Riley's discussion. The actual results at Colfax certainly show that the ashpit loss is less, that the clinker

[1] Closure to discussion prepared by Mr. Fairchild who presented the paper in the absence of Mr. Clarke.

is handled more easily, and that the operators prefer to operate the boiler with the preheater rather than those without the preheater.

One question that has been brought up is the variation in the draft loss on the gas side through the preheater. As noted in the paper, these figures given in the tabular form are not reliable on account of leakage, but other tests which were run with the air side of the preheater closed off indicate that a loss of about 0.25 in. exists.

No. 1912

BOILER-PLANT ECONOMICS

By N. E. Funk,[1] Philadelphia, Pa.
Member of the Society
and
Farley C. Ralston,[2] Philadelphia, Pa.
Non-Member

Noting the lack of adequate mathematical treatment in the literature on the subject, the authors have applied themselves to the determination of a rational formula for expressing boiler-plant performance. The paper shows how this formula may be applied, and also discusses the question of operating at minimum cost. The authors believe that with the analysis which they present considerably more rational practice can be followed, both in the matter of daily operation of existing boiler plants and in the design of new plants.

THE subject-matter of this paper falls naturally into three distinct parts; and it will be so presented.

2 Part I is devoted to the development of a rational formula for expressing boiler-plant performance, and to the determination, from boiler-efficiency guarantees or tests, of the values which the constants in the formula take for any given boiler.

3 In Part II the boiler formula is applied in solving the problems of operation at maximum boiler efficiency, at maximum boiler-room efficiency for a given load, and at maximum daily boiler-room efficiency.

4 In Part III the problem of operation at minimum total cost, including investment, operating and maintenance charges, is considered.

I — DEVELOPMENT OF A RATIONAL FORMULA FOR EXPRESSING BOILER-PLANT PERFORMANCE

5 The B.t.u. input per hour, the B.t.u. output per hour, and the efficiency of a boiler, all at any given rating, are connected by the following relation:

[1] Operating Engineer, Philadelphia Electric Company.
[2] Engineering Dept., The Philadelphia Electric Company.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

$$\text{B.t.u. input per hour} = \frac{\text{B.hp.} \times 33{,}479 \times x}{\text{Efficiency at output rate } x} \quad . \quad . \quad . \quad [1]$$

in which

B.hp. = nominal builders' boiler-horsepower rating of the boiler

x = boiler output rate; that is, the percentage rating at which the boiler is operating, expressed as a decimal; e.g., 100 per cent rating = 1.00 boiler-output rate

33,479 = number of B.t.u. in one boiler horsepower-hour.

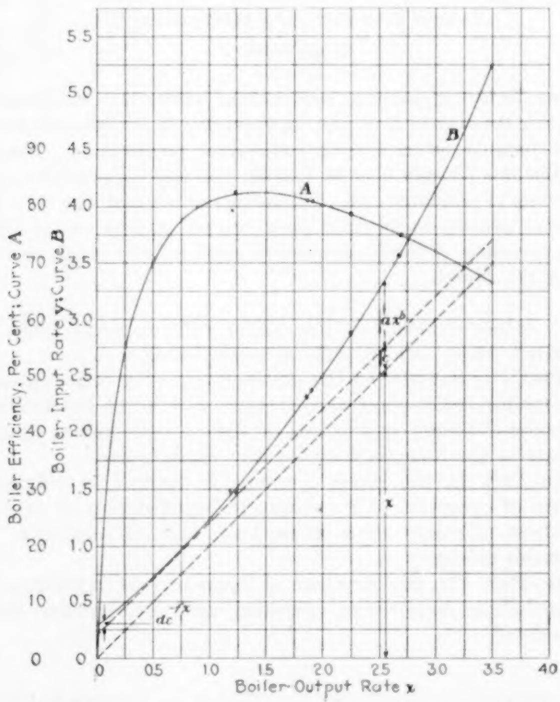

FIG. 1 METHOD OF CONSTRUCTING INPUT-OUTPUT CURVE FROM BOILER EFFICIENCY-OUTPUT RATE CURVE

Curve *A* is a typical boiler, furnace and economizer curve for a modern boiler equipped with an underfeed stoker. Curve *B* is shown as the sum of all the curves in Figs. 2 to 4, inclusive, into which it may be resolved.)

6 It will be noted that the right-hand member of Equation [1] is composed of a constant and a variable, the variable quantity, which will hereafter be known as the input rate, and designated by the letter y, being

$$y = \frac{x}{\text{Efficiency at output rate } x} \quad . \quad . \quad . \quad . \quad . \quad [2]$$

7 For the present purpose the input-output curve has the great advantage over the efficiency curve that it can be adequately represented by an algebraic equation of simple form, the constants of which are easily determined and have definite physical meanings.

8 Curve B, Fig. 1, has been obtained from curve A, which is a typical boiler, furnace and economizer efficiency curve for a modern boiler equipped with an underfeed stoker. Curve B may be represented by the formula

$$y = x + c + ax^b + d\epsilon^{-fx} + gx \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{3}]$$

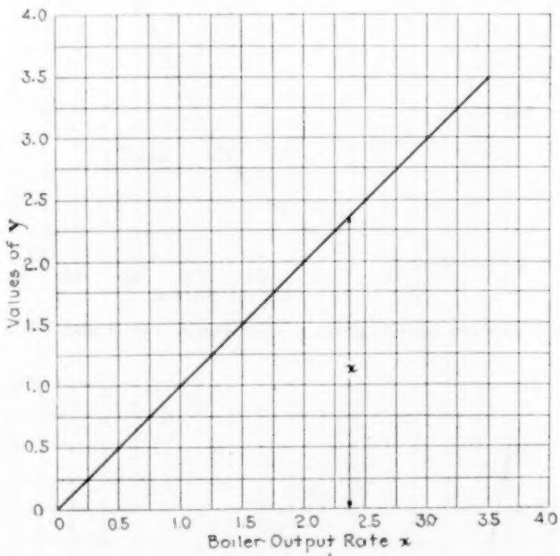

Fig. 2 Graph of Equation $y = x$

in which a, c, d, and g are coefficients, and b and f are exponents; these values for any given boiler plant are readily determined directly from boiler tests.

9 The physical meaning of each term of Equation [**3**] will now be separately shown and explained.

10 First, a boiler or a boiler plant, like any other apparatus for the conversion of energy, must have as one portion of its energy input an amount equal to the output, hence the first term of the equation, $y = x$ (see Fig. 2).

11 Second, the boiler or boiler plant has a constant loss, whether the boiler is carrying load or not. This loss is represented principally by the heat necessary to keep steam up at no load, by the no-load loss of fans, etc.; and all other losses which do not

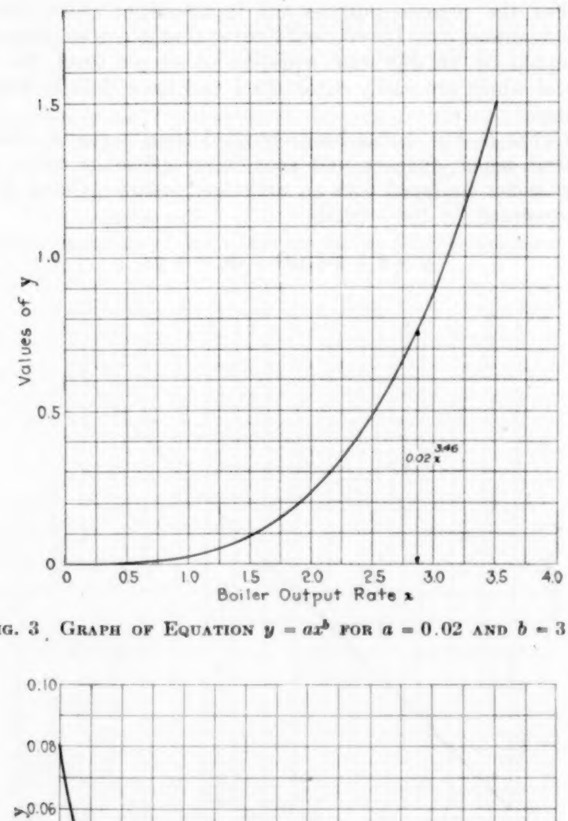

FIG. 3 GRAPH OF EQUATION $y = ax^b$ FOR $a = 0.02$ AND $b = 3.46$

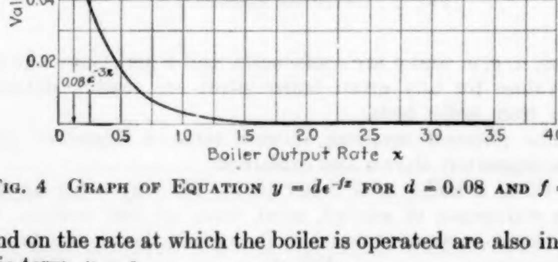

FIG. 4 GRAPH OF EQUATION $y = d\epsilon^{-fx}$ FOR $d = 0.08$ AND $f = 3$

depend on the rate at which the boiler is operated are also included in this term, $y = c$.

12 Third, a boiler or boiler plant has losses due to greater radiation, higher flue-gas temperatures, increased combustible in the ash, choking of the combustion chamber, increased fan power, etc., at the higher output rates, which increase in more than direct

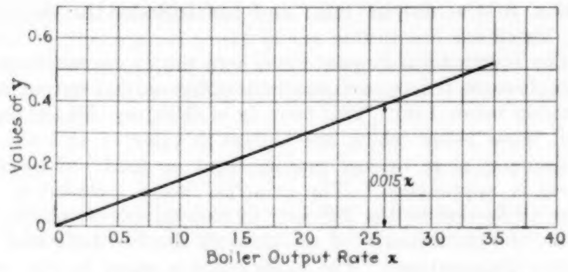

FIG. 5 GRAPH OF EQUATION $y = gx$ FOR $g = 0.015$

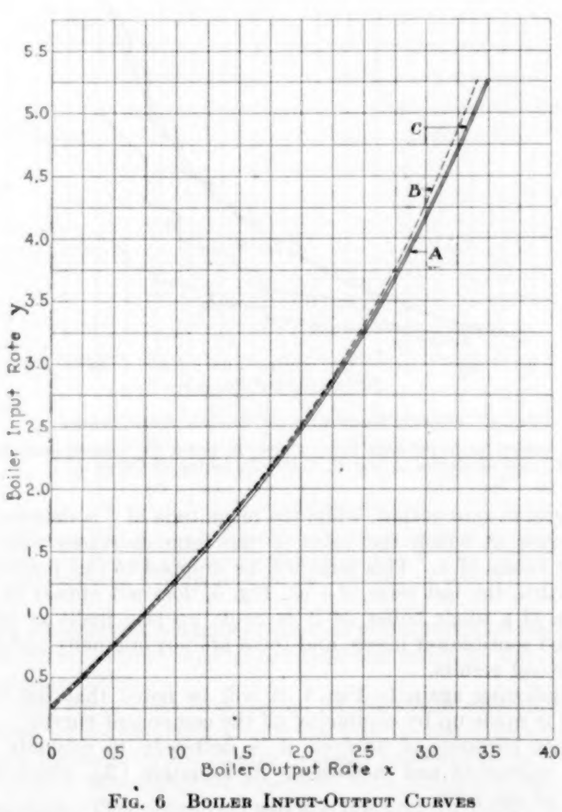

FIG. 6 BOILER INPUT-OUTPUT CURVES

(Curve *A* is plotted from data of tests. Curve *B* is Curve *A* corrected for steam consumption of boiler auxiliaries. Curve *C* is Curve *A* corrected for steam consumption of boiler auxiliaries and for variation of turbine steam rate with superheat.)

proportion to the output rate, and are included in the term $y = ax^b$, Fig. 3.

13 The fourth term is mentioned here simply to recognize its existence, because it has an appreciable value only at output rates considerably below 1.00. This term ($y = d\epsilon^{-fx}$, see Fig. 4) represents all those losses which are highest in value at no load, and which decrease more or less rapidly from no load until output rate 1.00 is approached.[1] The principal losses included in this term are those due to the difficulty of maintaining a uniform fire thickness; the low velocity of air through the fuel bed; and the low firebox temperatures. The coefficient d is equal to the value

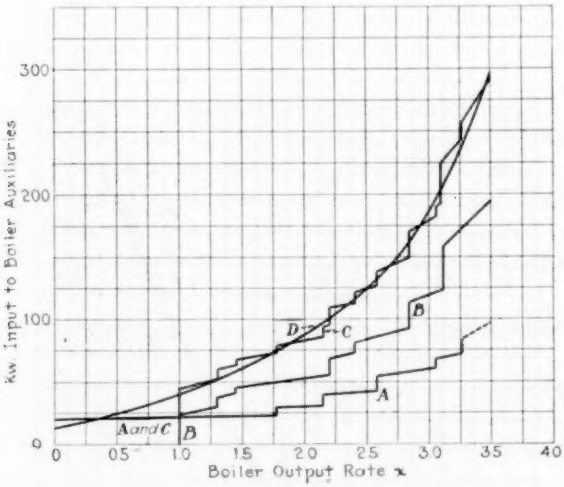

FIG. 7 POWER REQUIRED BY BOILER AUXILIARIES

Curve A, power for forced-draft fans; Curve B, power for induced-draft fans; Curve C, sum of Curves A and B; Curve D, fairing of Curve C.)

of the term at zero output, while the magnitude of f is determined by the rate at which the value of the term decreases with increasing values of x. This term will be dropped at this point.

14 Fifth, the last term, $y = gx$, Fig. 5, does not appear in the equation of a single boiler as it is made up principally of blowdown and soot-blower losses, and these are not originally included in boiler-test results.

15 Referring again to Fig. 1, it will be noted that the total curve B is made up by combining all the component curves.

16 For purposes of analysis it is necessary to evaluate the various exponents and coefficients in Equation [**3**], which now appears in the form

$$y = x + ax^b + c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{4}]$$

[1] ϵ is the base of the natural system of logarithms.

17 It should be remarked here that the values of a, b, and c are fixed by the original design of the boilers and the boiler plant; and once determined for any boiler plant they remain constant, except as careless operating methods result in poorer efficiency than should be obtained.

Determination of the Constants in the Boiler Formula

18 To determine for any given boiler or boiler plant the values of a, b, and c in the boiler formula, having given the boiler and

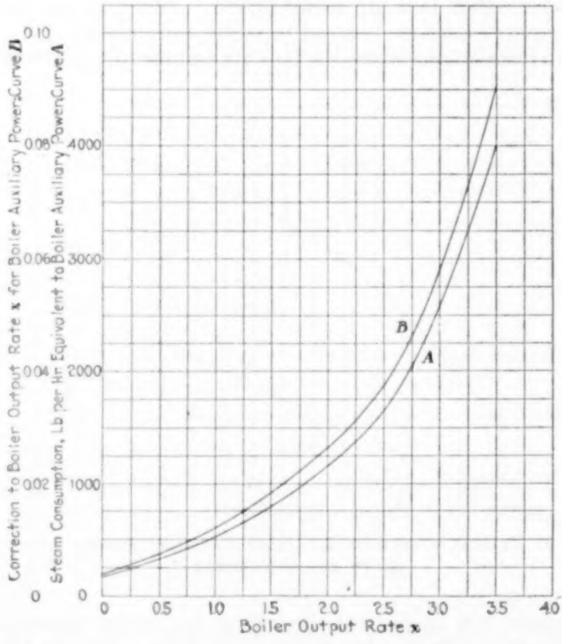

Fig. 8 Curve (A) Representing Steam Consumption of Main Turbines for Power Output of Curve D, Fig. 7, and Curve (B) of Corresponding Correction to be Applied in Fig. 6

furnace efficiency at a number of output rates, construct the input-output curve. Curve A, Fig. 6, shows the result of this process for a certain boiler.

19 This curve must be corrected for the steam consumption of the boiler auxiliaries; and also, if the load on the boilers consists of steam turbines, for the change in steam consumption of the turbines due to the change of superheat with change in the boiler output rate. The method of making the first correction is quite obvious, but the second may require explanation.

20 In Fig. 7, curve A represents the power for the forced-

draft fan, curve *B* that for the induced-draft fans, and curve *C* the sum of curves *A* and *B*. To facilitate plotting and computation, curve *D* has been drawn in place of *C* with its discontinuities due to the method of motor speed control.

21 In Fig. 8, curve *A* represents the steam consumption of the main turbines for the power output of curve *D*, Fig. 7, and curve *B* the corresponding correction to be applied in Fig. 6. Curve *B*, Fig. 8, is obtained from curve *A* by dividing the steam-consumption values of curve *A* by the number of pounds of steam delivered by the boiler at 100 per cent rating. In this installation the forced- and induced-draft fans are motor-driven. Therefore the electric power required must be reduced to steam consumption.

22 In installations where the fans are turbine-driven the steam

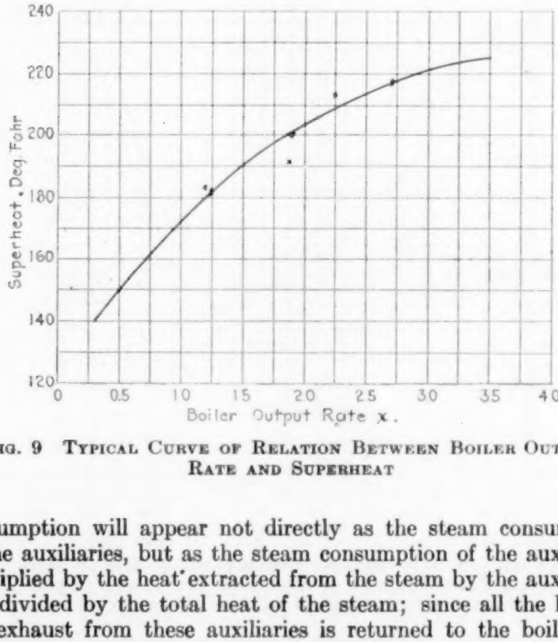

FIG. 9 TYPICAL CURVE OF RELATION BETWEEN BOILER OUTPUT RATE AND SUPERHEAT

consumption will appear not directly as the steam consumption of the auxiliaries, but as the steam consumption of the auxiliaries multiplied by the heat extracted from the steam by the auxiliaries and divided by the total heat of the steam; since all the heat in the exhaust from these auxiliaries is returned to the boiler and should not be considered in the total boiler output.

23 The power consumption of the stoker motor has been purposely neglected as being too small to be seen on the scale of Fig. 6.

24 To correct curve *A*, Fig. 6, for the auxiliary steam consumption, it must be remembered that at any boiler output rate the auxiliary steam consumption is subtracted from the available steam from the boiler at that output rate without reducing the coal consumption. The corrected curve is shown by the dotted line *B* in Fig. 6.

25 The superheat of a boiler varies more or less widely through its load range. With the superheat the heat content in B.t.u. per pound at steam condition above that of the boiler feedwater also varies, and consequently the number of pounds of steam per hour at 100 per cent of rating.

26 The steam rate of a turbine also varies considerably with the superheat of the steam supplied to it.

27 On account of this interrelation between the characteristics

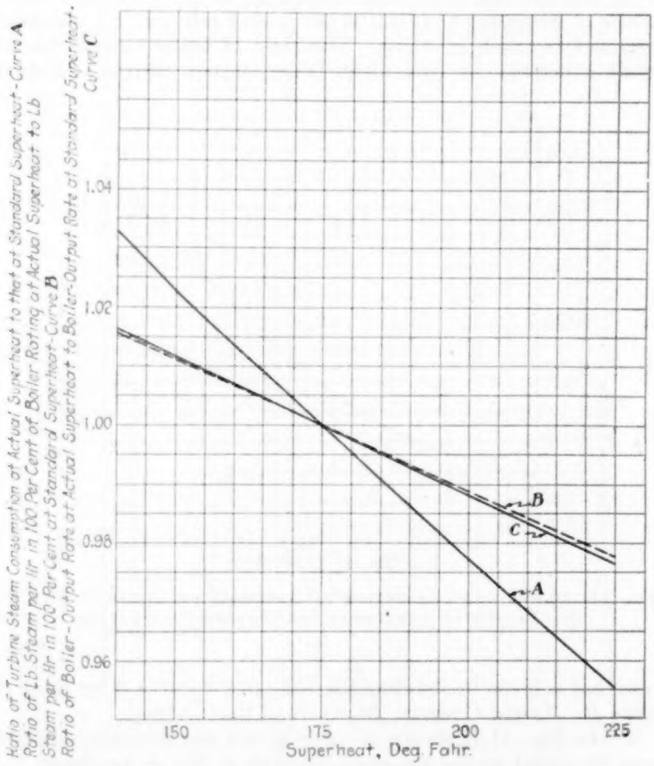

FIG. 10 VARIATION OF TURBINE STEAM CONSUMPTION AND EQUIVALENT BOILER OUTPUT RATE WITH SUPERHEAT

of the boiler and the turbine with respect to superheat, they must be considered as a unit.

28 A convenient method of accomplishing this is to correct the curve of actual boiler input-output at varying superheats to equivalent boiler input-output at a single value of superheat, preferably, for simplicity, that at which the turbine steam rates are specified; then, in all further calculations involving turbine steam rates, to use the rates specified at the standard superheat.

29 Fig. 9 is typical of the relation between superheat and boiler output rate.

30 In a particular turbine installation the guaranteed steam rates were specified at a standard superheat of 175 deg. fahr.

31 The ratio in which the steam consumption decreases with increasing superheat, as shown by curve *A*, Fig. 10, is not a clear gain of equivalent boiler output rate, because at the same time the number of pounds of steam per hour in 100 per cent of boiler rating is decreasing according to curve *B*, in which the number of pounds of steam per hour in 100 per cent of rating at the standard superheat is taken as unity. The ratio of boiler output rate at actual superheat to equivalent boiler output rate at standard

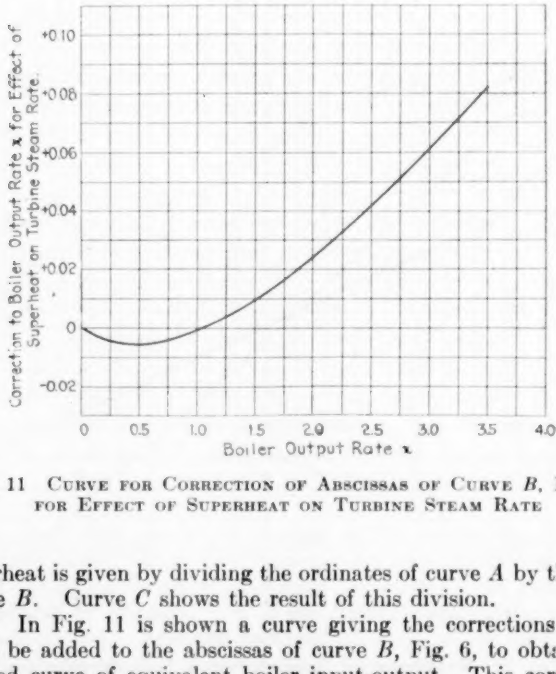

FIG. 11 CURVE FOR CORRECTION OF ABSCISSAS OF CURVE *B*, FIG. 6, FOR EFFECT OF SUPERHEAT ON TURBINE STEAM RATE

superheat is given by dividing the ordinates of curve *A* by those of curve *B*. Curve *C* shows the result of this division.

32 In Fig. 11 is shown a curve giving the corrections which must be added to the abscissas of curve *B*, Fig. 6, to obtain the desired curve of equivalent boiler input-output. This correction curve is developed by combining Fig. 9 and curve *C*, Fig. 10.

33 It might appear from the close agreement of the finally corrected curve with the uncorrected one, as if the labor of making the correction was not justified by the results of the process; but this agreement is incidental, depending on the data of the particular case, and is not at all inherent in the method.

34 The method of obtaining from curve *C*, Fig. 6, the values of the constants, *a*, *b*, and *c* in Equation [**4**] for boiler input-output will now be developed.

35 Subtract from each ordinate the corresponding abscissa. Plot these differences as ordinates against the same abscissas (Fig. 12). This gives the new formula:

$$z = y - x = ax^b + c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

in which z represents the total boiler losses at any given output rate.

36 From an inspection of the curve and the formula it will be noted that when x equals zero, ax^b also equals zero. Therefore

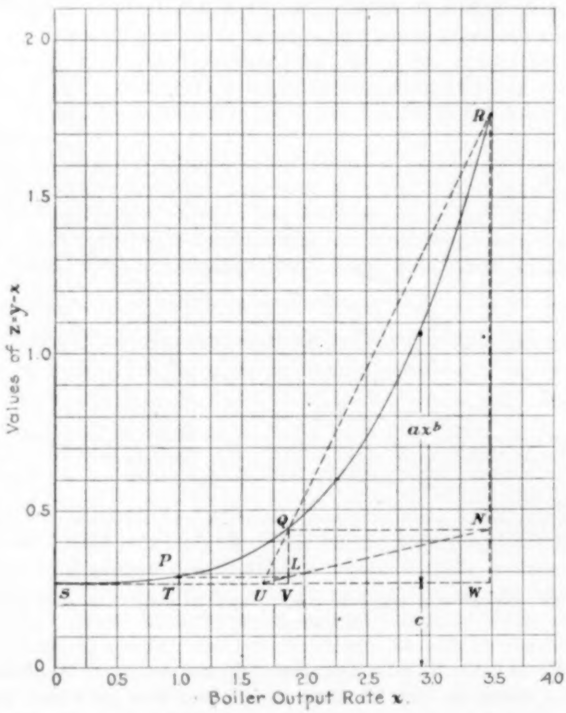

FIG. 12 GRAPHICAL METHOD OF DETERMINING THE VALUES OF THE CONSTANTS a AND c IN THE BOILER FORMULA EXPRESSED BY EQUATION [4]

$z_0 = c$. To determine c, select three points on the curve (P at x_1, z_1; Q at x_2, z_2; and R at x_3, z_3) such that

$$\frac{x_2}{x_1} = \frac{x_3}{x_2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

Then

$$\mathbf{c} = z_0 = \frac{z_2 - \dfrac{z_2 - z_1}{z_3 - z_2} z_3}{1 - \dfrac{z_2 - z_1}{z_3 - z_2}} \quad . \; . \; . \; . \; . \; . \; . \quad [7]$$

37 The proof of Equation [7] follows readily from the construction for the graphical determination of c. This construction is: Project a horizontal line from P until it intersects with the vertical line projected from the point Q at L; project a horizontal line from the point Q until it intersects a vertical line projected from the point R at N; draw a line through Q and R, and another through L and N; the point U at the intersection of these two lines is at the height $z_0 = c$.

38 Still using the last curve plotted in Fig. 12, or the equation $z = ax^b + c$, it will be noted that when $x = 1$, $ax^b = a$ and

$$z = a + c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

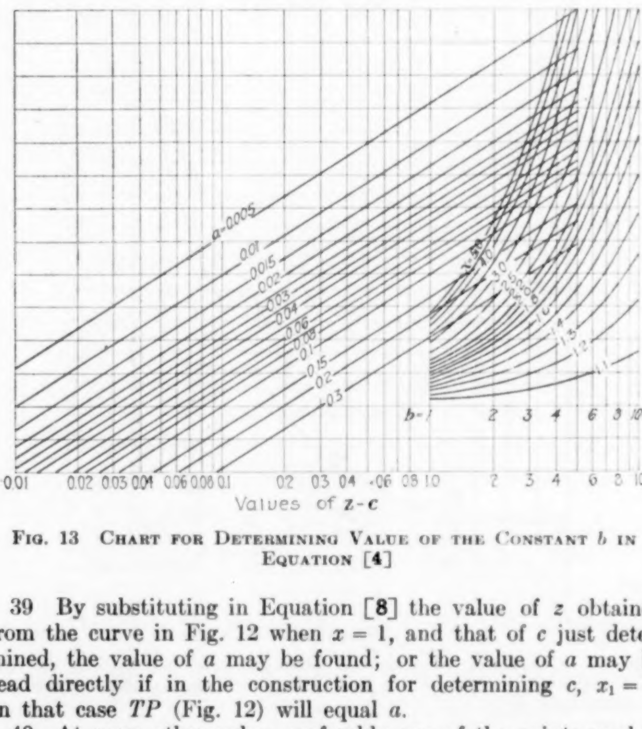

Fig. 13 Chart for Determining Value of the Constant b in Equation [4]

39 By substituting in Equation [8] the value of z obtained from the curve in Fig. 12 when $x = 1$, and that of c just determined, the value of a may be found; or the value of a may be read directly if in the construction for determining c, $x_1 = 1$. In that case TP (Fig. 12) will equal a.

40 At some other value, preferably one of the points used in determining c, substitute in the equation $z = ax^b + c$ the known values of c, a, and z, and solve for b. The values of a and b can also be determined graphically, using logarithmic cross-section paper, by placing Equation [5] in the form

$$z - c = ax^b$$

41 An alternative graphic solution for the value of b is given in Fig. 13. To use this chart, enter at the selected value of $(z - c)$

on the lower scale; proceed vertically to the line representing the value of a as already determined, then horizontally to the curve representing the value of x which corresponds to the chosen value of $(z - c)$. The abscissa of the point thus found represents the desired value of b, according to the scale on the axis so marked.

42 The procedure in application is as follows: Obtain the input-output curve by dividing the output rate by the efficiency at that output rate. Correct the input-output curve for boiler auxiliaries and superheat. Obtain curve for Equation [**5**] by subtracting the output rate from the input rate for each output rate point. Select on this curve three points whose abscissas are in the proportion given in Equation [**6**]. Solve for c by the use of Equation [**7**]. Solve for a by using Equation [**8**] and the value of z obtained from the curve when $x = 1$. Solve for b by the use of Fig. 13.

43 All of the constants in Equation [**4**] have now been determined, and the numerical values of a, b, and c can be substituted for the literal ones. Data obtained from an actual boiler installation have been used in preparing the foregoing charts; and for this installation the constants have been determined as follows:

$$a = 0.02;\ b = 3.46;\ c = 0.27$$

Equation [**4**] can therefore now be written (for this installation)

$$y = x + 0.02x^{3.46} + 0.27$$

II — APPLICATION OF FORMULA IN ANALYSIS OF BOILER-PLANT OPERATION

44 The analysis to follow is strictly limited in its application to boiler plants in which all of the boilers have equal ratings and similar performance characteristics. It is further assumed that any station load is divided equally among the boilers which are steaming. These limitations are not unwarranted in the case of many modern plants; and without them, as will be shown later, the analysis would become unduly complex.

The following symbols will be used in this analysis:

K = number of tons (2000 lb.) of coal to produce one boiler hp-hr. at 100 per cent efficiency
K_B = number of tons (2000 lb.) of coal to bank one boiler hp. for one hour
H = nominal rating of each boiler in boiler hp.
f = load factor of the station load (as a decimal; 100 per cent = 1)
L = station load in boiler hp.
N = number of boilers steaming at the load L
h = number of hours' duration of the load L
x = boiler output rate at the station load L and with N boilers steaming

y = boiler input rate corresponding to the boiler output rate x
E = boiler efficiency (expressed as a decimal; 100 per cent = 1) at the boiler output rate x.
T_S = number of tons (2000 lb.) of coal burned per steaming hour per boiler at the boiler output rate x.

45 In particular, L_1, N_1, h_1, x_1, y_1, E_1, and T_{S1} refer to the maximum station load of the day. L_2, L_3, etc.; N_2, N_3, etc.; h_2, h_3, etc.; x_2, x_3, etc.; y_2, y_3, etc.; E_2, E_3, etc.; T_{S2}, T_{S3}, etc., refer to loads other than the daily maximum.

46 It is assumed that at the time of the daily maximum load no boilers are banked. Therefore N_2, N_3, etc., may be equal to N_1, but cannot be greater, and

$N_1 - N_2$, $N_1 - N_3$, etc. = number of banked boilers at loads L_2, L_3, etc.
T_B = number of tons (2000 lb.) of coal burned per banked hour per boiler
T_U = number of tons (2000 lb.) of coal burned per *hour* by the boiler room, including all steaming and banked boilers
T_T = number of tons (2000 lb.) of coal burned per *day* by the boiler room, including all steaming and banked boilers.

47 The value of K is equal to the number of B.t.u. in one boiler horsepower divided by the number of B.t.u. per ton of coal. For coal of 13,500 B.t.u. per lb., $K = 0.00124$.

48 Tests made on many types of underfeed stokers show that about 6750 B.t.u. are required per boiler hp-hr. to bank a boiler, so that $K_B = 0.00025$.

49 The ratio $K_B/K = 0.202$ will be used frequently in the following development. It will be noted that this ratio is a constant regardless of the B.t.u. value of the particular coal used. Further,

$$L = NHx \quad [9]$$

$$T_S = HKy \quad [10]$$

$$T_B = HK_B \quad [11]$$

$$T_U = NT_S + (N_1 - N)T_B = NHKy + (N_1 - N)HK_B \quad [12]$$

Determination of Conditions for Maximum Boiler Efficiency

50 To determine the output rate of a single boiler for maximum boiler efficiency, the reciprocal of the efficiency, $1/E$, has been used and differentiated for a minimum.

$$\frac{1}{E} = \frac{y}{x} = \frac{x + ax^b + c}{x} = 1 + ax^{b-1} + cx^{-1}$$

Differentiating, equating to zero, substituting x_E (where x_E is the output rate for maximum boiler efficiency), and solving,

$$x_E = \left[\frac{c}{a(b-1)}\right]^{\frac{1}{b}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [13]$$

51 A convenient graphical solution of Equation [**13**] is shown in Fig. 14. The chart is used by entering at the bottom of the left

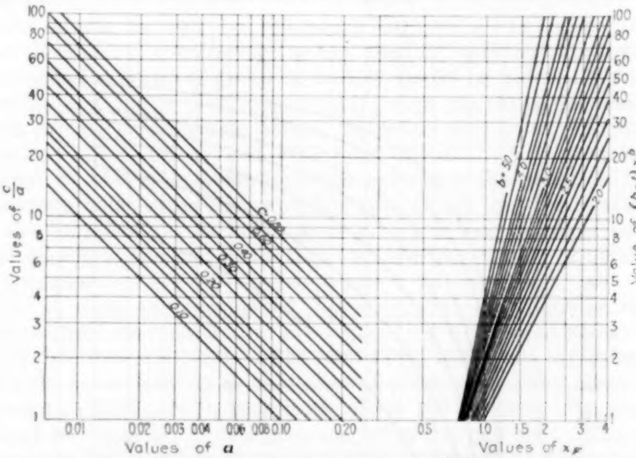

FIG. 14 GRAPHICAL SOLUTION OF EQUATION [**13**] FOR THE BOILER OUTPUT RATE FOR MAXIMUM BOILER EFFICIENCY

side at the point representing the value of a; proceeding vertically to the inclined line representing the value of c, then horizontally to the right to the inclined line representing the value of b. The abscissa at this point represents x_E, the output rate for maximum boiler efficiency.

DETERMINATION OF CONDITIONS FOR MAXIMUM BOILER-ROOM EFFICIENCY FOR A GIVEN LOAD

52 To determine, for any given load, the boiler output rate for maximum efficiency of the boiler room as a whole (including banking), or in other words, the output rate at which boilers should be banked on a decreasing load and put on the line on an increasing load, it is necessary, after making appropriate substitutions, to differentiate, with respect to x, Equation [**12**] for the hourly boiler-room coal consumption, and to solve for a minimum.

$$T_U = \frac{LKy}{x} + N_1HK_B - \frac{LK_B}{x} = LK\ (1 + ax^{b-1} + cx^{-1})$$
$$+ N_1HK_B - LK_Bx^{-1}$$

53 Differentiating, equating to zero, substituting x_F (where x_F = boiler output rate for maximum boiler-room efficiency) for x and solving,

$$x_F = \left[\frac{c - \frac{K_B}{K}}{a(b-1)}\right]^{\frac{1}{b}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{14}]$$

54 It must be understood that Equation [**14**] applies only to station loads small enough that at least one boiler is banked.

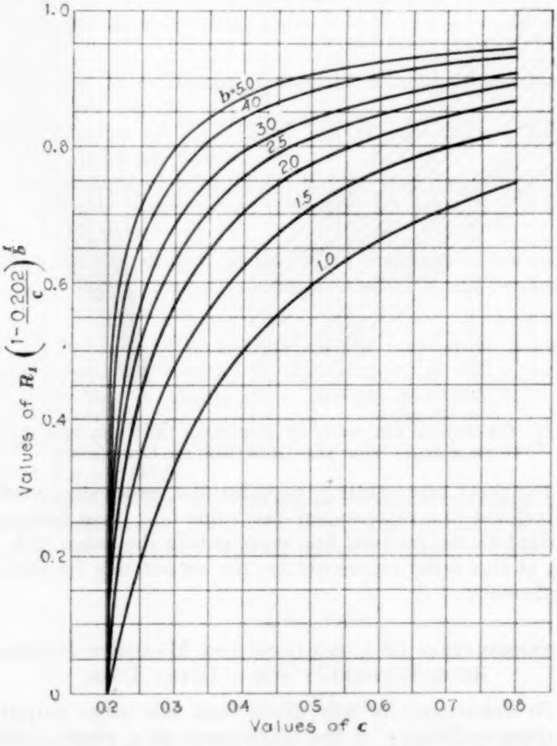

Fig. 15 Graphical Solution of Equation [**16**]

For all loads such that no banking is required, $N = N_1$, $N_1 - N = 0$, and the term K_B/K drops out, leaving Equation [**13**].

55 The similarity between Equation [**13**], for the output rate giving maximum efficiency of a single boiler, and Equation [**14**], for the output rate at which boilers should be banked, can be utilized as follows: Equation [**14**] can be written in the form

$$x_F = \left[\frac{c}{a(b-1)}\right]^{\frac{1}{b}} \left[\frac{c - \frac{K_B}{K}}{c}\right]^{\frac{1}{b}} = x_E R_1 \quad . \; . \quad [15]$$

in which

$$R_1 = \left[1 - \frac{0.202}{c}\right]^{\frac{1}{b}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

56 A graphical solution of Equation [**16**] will be found in Fig. 15.

57 It will be seen by comparing Equations [**13**] and [**15**] that the effect of considering the coal used for banking is to lower the rating to which a group of boilers should be reduced before banking any one of them considerably below the rating for maximum efficiency of a single boiler. An interesting point, and one which is not at all obvious except by this analysis, is that the *number* of boilers in operation does not enter the equation for maximum boiler-room efficiency explicitly at all; but the relation of the constants in the boiler formula determines the most economical *output rate*, and this is true whatever the number of boilers in operation.

58 It must be remembered, however, that the shape of the input-output curve from 100 per cent rating to 0 is not satisfactorily determined by tests. The term $d\epsilon^{-fx}$, which was eliminated from the boiler formula as used above for this reason and because it becomes negligible in magnitude above 100 per cent of rating, cannot be neglected at ratings much below 100 per cent. Therefore, if the use of certain values of a, b, and c is found to give values of x_F lower than 80 per cent of rating, these results should not be accepted without more careful investigation.

59 Fig. 16 shows the total coal per hour, T_U, required for station loads of 10,000 and 15,000 boiler hp. when the number of steaming boilers is varied. For some distance on either side of the minimum point, each of these curves is quite flat. This indicates that the actual output rate at which boilers are banked can vary over a considerable range without increasing the hourly coal consumption appreciably above that at the theoretically most efficient output rate x_F.

60 In the foregoing discussion it has been assumed that L is a block load lasting for one hour, and that T_S, T_B, and T_U represent the quantities of coal burned in one hour. The entire reasoning may be applied with equal force to a block load of any duration h.

61 The coal consumption during any load block is then obviously

$$hT_U = h\left[NT_S + (N_1 - N)T_B\right] \quad . \; . \; . \; . \; . \quad [17]$$

Determination of Conditions for Maximum Daily Boiler-Room Efficiency

62 In the general case it is considered that any daily load curve consists of n rectangular load steps, n being any number which may be required to represent properly the actual load

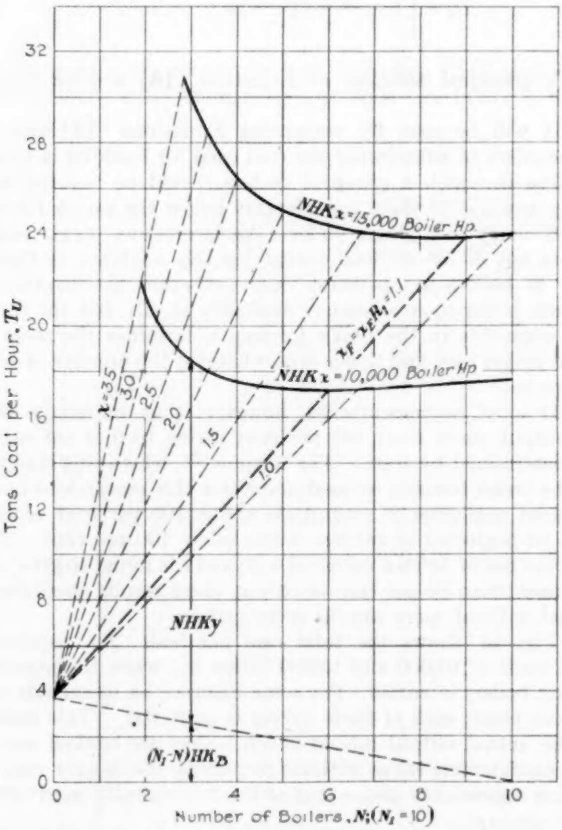

FIG. 16 VARIATION OF TOTAL HOURLY COAL CONSUMPTION FOR STATION LOADS OF 10,000 AND 15,000 BOILER HP. WITH NUMBER OF STEAMING BOILERS

curve. Each of these n load steps may be of any height and of any duration, as is illustrated in Fig. 17.

63 The load steps are numbered consecutively from the highest downward $L_1, L_2, L_3 \ldots\ldots L_n$. For each load step the number of boilers steaming (N), the boiler output rate (x), the boiler-input rate (y), and the duration (h) are designated by the same subscript as the corresponding station load (L).

64 Preliminary to the derivation of the formula for operation at maximum daily boiler-room efficiency, certain relations will be stated for use in the derivation. By Equation [**19**]

$$N_1H = \frac{L_1}{x_1},\ N_2H = \frac{L_2}{x_2},\ N_3H = \frac{L_3}{x_3}\ .\ .\ .\ .\ .\ N_nH = \frac{L_n}{x_n}\ .\ .\quad [\mathbf{18}]$$

$$\frac{N_2}{N_1} = \frac{L_2x_1}{L_1x_2},\ \frac{N_3}{N_1} = \frac{L_3x_1}{L_1x_3}\ .\ .\ .\ .\ .\ \frac{N_n}{N_1} = \frac{L_nx_1}{L_1x_n}\ .\ .\ .\ .\ .\ .\quad [\mathbf{19}]$$

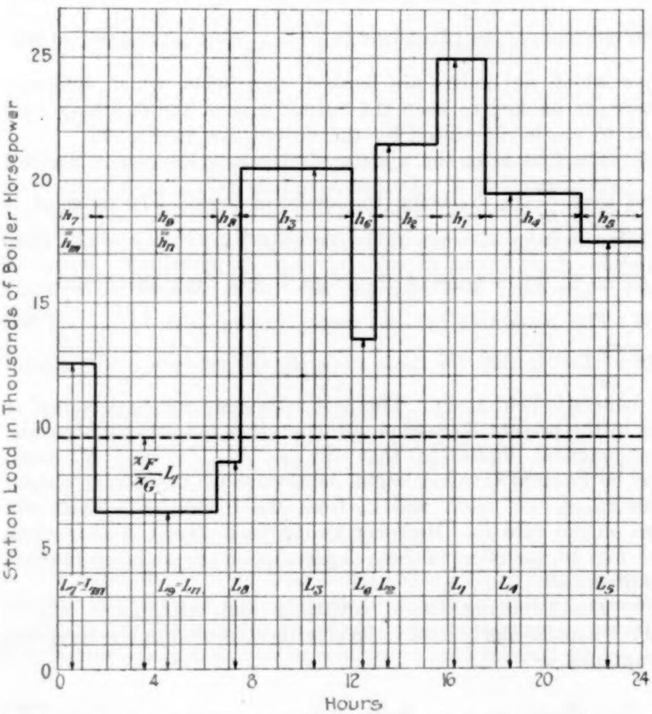

Fig. 17 Typical Daily Boiler-Plant Load Curve

$$x_2 = \frac{L_2N_1}{L_1N_2}x_1,\ x_3 = \frac{L_3N_1}{L_1N_3}x_1\ .\ .\ .\ .\ .\ x_n = \frac{L_nN_1}{L_1N_n}x_1\ .\ .\ .\quad [\mathbf{20}]$$

65 In the preceding derivation of x_F (Equation [**15**]), N_1 has been assumed as known, and it has been shown that the minimum boiler-room coal consumption for any plant load block is determined when N_1 is known. However, in general, each value of N_1 results in a different value for this minimum; so that for minimum daily coal consumption on a given load curve there is a most favorable value of N_1 which must be found. Our problem,

then, is to determine this best value for N_1; or, what is equally useful, the best value of x_1.

66 For each load step except the first it is obvious that there are two alternatives: banking may be required, or it may not. It has previously been shown that the output rate of a group of boilers must be reduced to x_P before banking any of them, in order to obtain maximum boiler-room efficiency with any given load. At this output rate and with no boilers banked (whether or not N_1 has been properly chosen to give maximum daily boiler-room efficiency), the plant load is $\frac{x_P}{x_1} L_1$. This plant load, at and above which no boilers are banked, and below which sufficient boilers are banked to keep the output rate of the steaming ones equal to x_P, is illustrated by the dotted line in Fig. 17.

67 The first m of the n steps of the daily load curve are not lower than $\frac{x_P}{x_1} L_1$, L_m being the lowest of these. The range of m in value is from 1, when L_1 is the only load step not less than $\frac{x_P}{x_1} L_1$, to n, when none of the steps is less than that value.

68 The remaining $(n - m)$ steps, starting with $L_{(m+1)}$, are less than $\frac{x_P}{x_1} L_1$; so that for these steps the output rate of the steaming boilers is held at x_P. This last statement will be assumed as true in the present discussion, although it is not strictly accurate. The practical limitation that $N_{(m+1)}$ . . . . . and N_n must each be integral involves slight departures of the actual output rates $x_{(m+1)}$, ... and x_n from the theoretically most efficient output rate x_P. However, as has been shown in connection with Fig. 16, the effect of these departures on the coal consumption is quite small.

69 The following set of equations and relations — **[21]** to **[29]** — express algebraically the conditions stated in the foregoing paragraphs.

$$N_2 = N_3 = \ldots\ldots = N_m = N_1 \quad \ldots\ldots\ldots\ldots \quad [21]$$

$$N_1 - N_2 = N_1 - N_3 = \ldots\ldots = N_1 - N_m = 0 \quad \ldots \quad [22]$$

$$\frac{N_2}{N_1} = \frac{N_3}{N_1} = \ldots\ldots = \frac{N_m}{N_1} = 1 \quad \ldots\ldots\ldots\ldots \quad [23]$$

$$x_2 = \frac{L_2 N_1}{L_1 N_2} x_1 = \frac{L_2}{L_1} x_1, \quad x_3 = \frac{L_3}{L_1} x_1 \ldots\ldots x_m = \frac{L_m}{L_1} x_1 \quad . \quad [24]$$

$$y_1 = x_1 + a x_1^b + c, \quad y_2 = x_2 + a x_2^b + c = \left(\frac{L_2}{L_1}\right) x_1 + a \left(\frac{L_2}{L_1}\right)^b x_1^b + c,$$

$$y_3 = \left(\frac{L_3}{L_1}\right)x_1 + a\left(\frac{L_3}{L_1}\right)^b x^b_1 + c \ldots\ldots y_m = \left(\frac{L_m}{L_1}\right)x_1$$

$$+ a\left(\frac{L_m}{L_1}\right)^b x^b_1 + c \ldots\ldots\ldots\ldots \quad [25]$$

$$x_{(m+1)} = \ldots\ldots = x_n = x_F \ldots\ldots\ldots\ldots \quad [26]$$

$$y_{(m+1)} = \ldots\ldots = y_n = x_F + ax_F^b + c \ldots\ldots\ldots \quad [27]$$

$$\frac{N_{(m+1)}}{N_1} = \frac{L_{(m+1)}x_1}{L_1x_{(m+1)}} = \frac{L_{(m+1)}x_1}{L_1x_F} \ldots\ldots \frac{N_n}{N_1} = \frac{L_nx_1}{L_1x_F} \ldots\ldots \quad [28]$$

$$L_m \nless \frac{x_F}{x_1}L_1,\ L_{(m+1)} < \frac{x_F}{x_1}L_1 \ldots\ldots\ldots\ldots \quad [29]$$

70 The procedure for determining the value of x_1 resulting in maximum daily boiler-room efficiency is similar to that for determining x for maximum boiler-room efficiency with a block load; namely, setting up the equation for total daily coal consumption, expressing all the variables in terms of x_1, differentiating with respect to x_1, and solving for a minimum. From Equation [**17**]

$$\begin{aligned}T_T &= h_1T_{U1} + h_2T_{U2} + h_3T_{U3} + \ldots\ldots + h_mT_{Um} + h_{(m+1)}\ T_{U(m+1)} \\ &\quad + \ldots\ldots + h_nT_{Un} \\ &= h_1N_1HKy_1 + h_2N_2HKy_2 + h_3N_3HKy_3 + \ldots\ldots \\ &\quad + h_mN_mHKy_m + h_{(m+1)}N_{(m+1)}HKy_{(m+1)} + \ldots\ldots \\ &\quad + h_nN_nHKy_n + h_2(N_1 - N_2)HK_B + h_3(N_1 - N_3)HK_B \\ &\quad + \ldots\ldots + h_m(N_1 - N_m)HK_B + h_{(m+1)}(N_1 - N_{(m+1)})HK_B \\ &\quad + \ldots\ldots + h_n(N_1 - N_n)HK_B \ldots\ldots\ldots \quad [30]\end{aligned}$$

71 Substituting from Equations [**18**] to [**29**], collecting terms, clearing of fractions with respect to x_1, differentiating, equating to zero, substituting x_G for x_1 (where x_G is the value of x_1 resulting in maximum daily boiler-room efficiency), and solving, gives

$$x_G = \left[\frac{c}{a(b-1)}\right]^{\frac{1}{b}} \times \left[\frac{(h_1+h_2+h_3+\ldots\ldots+h_m)\left(1 - \frac{K_B}{Kc}\right) + \frac{24K_B}{Kc}}{h_1+h_2\left(\frac{L_2}{L_1}\right)^b + h_3\left(\frac{L_3}{L_1}\right)^b + \ldots\ldots + h_m\left(\frac{L_m}{L_1}\right)^b}\right]^{\frac{1}{b}} = x_BR_2 \quad [31]$$

in which

$$R_2 = \left[\frac{(h_1 + h_2 + h_3 + \ldots\ldots + h_m)\left(1 - \frac{0.202}{c}\right) + \frac{4.848}{c}}{h_1 + h_2\left(\frac{L_2}{L_1}\right)^b + h_3\left(\frac{L_3}{L_1}\right)^b + \ldots\ldots + h_m\left(\frac{L_m}{L_1}\right)^b}\right]^{\frac{1}{b}} \quad [32]$$

72 In preparing a graphical solution for Equation [**32**], it would have been possible to lay out a single chart on which to perform the entire operation graphically. This chart, however, would of necessity have been large, complicated, and quite cumbersome in use. Actually the process has been divided into two simple graphical steps, with two arithmetical summations, at a great gain in convenience.

73 Fig. 18, which is the first graphical step, has been laid out for the determination of the value of $h_p(L_p/L_1)^b$, in which L_p is the height of any load step and h_p the corresponding duration. To use this chart, enter the left-hand scale at the given value of

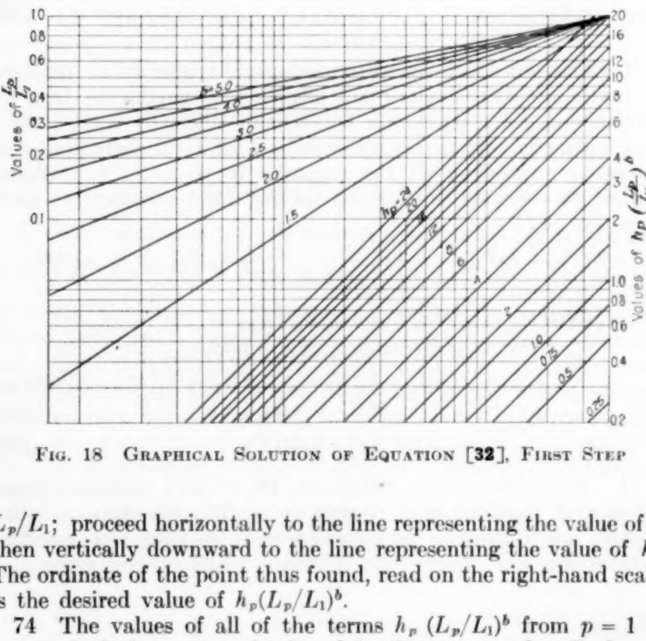

FIG. 18 GRAPHICAL SOLUTION OF EQUATION [**32**], FIRST STEP

L_p/L_1; proceed horizontally to the line representing the value of b; then vertically downward to the line representing the value of h_p. The ordinate of the point thus found, read on the right-hand scale, is the desired value of $h_p(L_p/L_1)^b$.

74 The values of all of the terms $h_p\ (L_p/L_1)^b$ from $p = 1$ to $p = m$, inclusive, are to be found in the manner described, and summated arithmetically. The sum of the terms gives the value of the expression

$$h_1 + h_2\left(\frac{L_2}{L_1}\right)^b + h_3\left(\frac{L_3}{L_1}\right)^b + \ldots\ldots + h_m\left(\frac{L_m}{L_1}\right)^b$$

which forms the denominator of the right-hand member of Equation [**32**]. This expression can be written, in more condensed notation, as

$$\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]$$

75 The value of the expression $h_1 + h_2 + h_3 + \ldots\ldots + h_m$, which occurs in the numerator of the right-hand member of Equation [**32**], is to be found by arithmetical summation of all of the terms h_p from $p = 1$ to $p = m$, inclusive. This expression will be similarly condensed to

$$\sum_{p=1}^{p=m}\Big[h_p\Big]$$

76 When Equation [**32**] is rewritten, using the condensed notation, thus:

$$R_2 = \left\{\frac{\left(1 - \frac{0.202}{c}\right)\sum_{p=1}^{p=m}\Big[h_p\Big] + \frac{4.848}{c}}{\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]}\right\}^{\frac{1}{b}} \quad \ldots\ldots \quad [\mathbf{32a}]$$

it is seen that the number of independent variables affecting the value of R_2 has been reduced to four:

$$\sum_{p=1}^{p=m}\Big[h_p\Big],\ \sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right],$$

b, and c.

77 Fig. 19 completes the solution. To use this chart, enter at the given value of $\sum_{p=1}^{p=m}\Big[h_p\Big]$ on the lower scale; proceed vertically to the line representing the value of c; then horizontally (left or right as may be required) to the line representing the value of $\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]$; then vertically upward to the line representing the value of b. The ordinate of the point thus found, read on the left-hand scale, is the value of R_2.

78 When R_2 has been determined for any given boiler plant and any given load curve, the value of x_1 for maximum daily boiler-room efficiency follows directly from the value of x_E already found from [**13**]. It is seen that for accuracy in determining the value of R_2, and consequently that of x_G, it is necessary to have used as m the true number of load steps, the lowest of which is not less than $(x_F/x_1)L_1$.

79 In the case of many actual load curves the number of such load steps is evident on inspection; for many others it is not. A simple test is available for checking the correctness of the value of m which has been used in Equation [**32**].

80 When, using the chosen value of m, R_2 has been determined, and from it x_G, substitute x_G as thus found for x_1 in the relations

$$L_m \nless \frac{x_F}{x_1} L_1,\ L_{(m+1)} < \frac{x_F}{x_1} L_1 \quad \ldots\ldots \quad [\mathbf{29}]$$

81 If $L_m \nless \frac{x_F}{x_G} L_1$ and $L_{(m+1)} < \frac{x_F}{x_G} L_1$, the correct value of m has been used. If $L_{(m+1)} \nless \frac{x_F}{x_G} L_1$, m has been chosen too small; if $L_m < \frac{x_F}{x_G} L_1$, m is too large. In either case after the value of m has been adjusted in the appropriate direction and new values determined for R_2 and x_G, the test should be re-

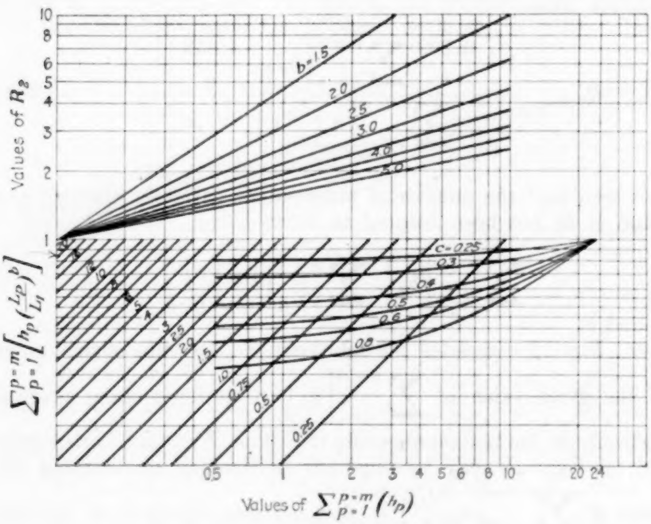

FIG. 19 GRAPHICAL SOLUTION OF EQUATION [**32**], FINAL STEP

peated to insure that the proper amount of adjustment has been made.

82 Substituting x_G for x_1 in Equation [**9**], thus:

$$N_1 = \frac{L_1}{Hx_G}$$

will give, in general, either an integral or a non-integral *theoretical* best value for N_1. But only the integral values are physically possible; therefore, if a non-integral value is found, the nearest integer must be used as the *actual* best value of N_1.

83 This adjustment of N_1 to an integral value involves a compensating departure of the actual best value of x_1 from the theoretical best value x_G; for, since L_1 and H are constants, the product N_1x_1 must be constant.

84 Substituting this actual value of x_1 in Equations [**24**],

$$x_2 = \left(\frac{L_2}{L_1}\right) x_1, \quad x_3 = \left(\frac{L_3}{L_1}\right) x_1 \; . \; . \; . \; . \; . \; x_m = \left(\frac{L_m}{L_1}\right) x_1$$

85 The theoretical value of $x_{(m+1)}$ . . . . . and x_n is x_F (Eq. [**26**]); that of $N_{(m+1)}$ is $\dfrac{N_1 L_{(m+1)} x_1}{L_1 x_F}$ . . . . . and that of N_n is $\dfrac{N_1 L_n x_1}{L_1 x_F}$ (Eq. [**28**]). But here the same physical limitation exists as in the case of N_1; that is, the nearest integers to the theoretical values of $N_{(m+1)}$ . . . . . and N_n must be used as the actual values. The actual values of $x_{(m+1)}$ . . . . . and x_n will then be given by Equations [**20**]:

$$x_{(m+1)} = \frac{L_{(m+1)} N_1}{L_1 N_{(m+1)}} x_1 \; . \; . \; . \; . \; . \; x_n = \frac{L_n N_1}{L_1 N_n} x_1$$

86 With all of the values x_1, x_2, x_3 . . . . . and x_n determined, the corresponding values y_1, y_2, y_3 . . . . . and y_n may be read from curve C, Fig. 6.

87 Sufficient data are now available to evaluate completely T_T in Equation [**30**]. For convenience in solving for T_T, this equation may be written, substituting 0.202 for K_B/K and clearing of fractions,

$$\begin{aligned} T_T = HK\{&N_1(h_1 y_1 + h_2 y_2 + h_3 y_3 + \; . \; . \; . \; . \; . \; + h_m y_m) \\ &+ (h_{(m+1)} N_{(m+1)} y_{(m+1)} + \; . \; . \; . \; . \; . \; + h_n N_n y_n) \\ &+ 0.202[N_1(h_{(m+1)} + \; . \; . \; . \; . \; . \; + h_n) - (h_{(m+1)} N_{(m+1)} \\ &+ \; . \; . \; . \; . \; . \; + h_n N_n)]\} \qquad [\mathbf{30a}] \end{aligned}$$

The resulting value of T_T is the desired minimum daily coal consumption for the given load curve.

Consideration of the Limitations Heretofore Imposed and Outline of Procedure When These Limitations Do Not Apply

88 Under the limitations specified at the beginning of Part II, the foregoing complete analysis results in final formulas which are exact, yet simple and workable.

89 Consider the most elementary case in which these limitations do not hold; that is, a station load of any given value, carried by two steaming boilers, which may or may not have equal ratings and similar characteristics. Suppose that the corrected input-output curve of each boiler has been drawn and the respective constants determined. Then it will be evident that before any calculation can be made of the minimum combined coal consumption of steaming and banked boilers, or of the

minimum daily boiler-room coal consumption, the proper distribution of the given station load between the two boilers for minimum steaming coal consumption is absolutely essential.

90 If the two boilers are of equal ratings but of dissimilar characteristics, a proper comparison between them may be obtained from the input-rate-output-rate curves without change;

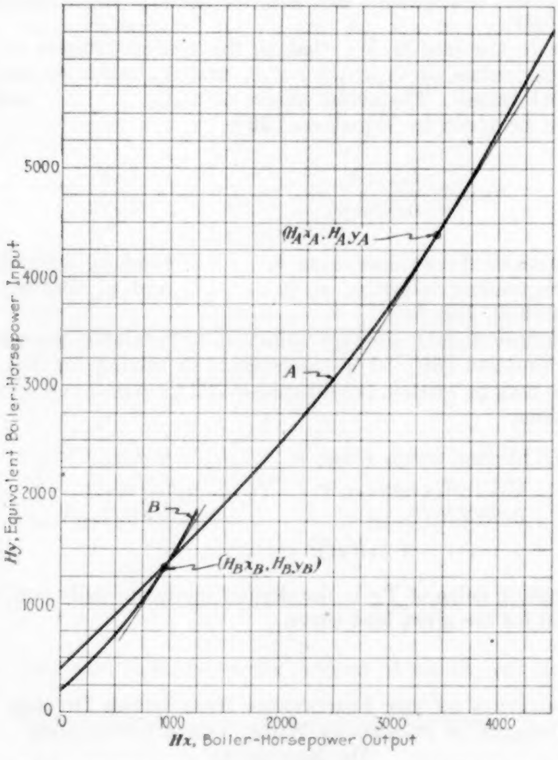

FIG. 20 PLOTTING OF PERFORMANCE CURVES OF TWO BOILERS FOR PURPOSE OF DETERMINING MINIMUM COMBINED INPUT

but if the ratings are unequal, the input-rate-output-rate curves must be redrawn so that the abscissas of both curves are measured on one scale and the ordinates of both curves also on one scale. It is not necessary that the ordinates and the abscissas be measured on the same scale; the abscissas may be in B.t.u. per hour or in boiler horsepower, while the ordinates may be in B.t.u. per hour, equivalent boiler horsepower, or tons or pounds per hour of coal of specified heating value.

91 On Fig. 20 are shown two boiler-performance curves, *A*

and B, redrawn with both abscissas and ordinates to the scale of 1000 boiler hp.

92 Whatever the output of each boiler may be, let a tangent be drawn to its performance curve at the point whose abscissa represents the output. The corresponding ordinates then express, to their scale, the inputs to the boilers; the sum of the ordinates the combined input; and the sum of the abscissas the combined output.

93 The condition which makes the combined input a minimum for a given combined output is that the tangents to the two curves at the respective output points be parallel; that is, that the curves have equal slopes. It is easy to see that only under this condition of load distribution will an infinitesimal increase in the output of either boiler, accompanied by a compensating decrease in the output of the other (and consequently no change in the combined output), result in similarly compensating changes in the individual inputs, and hence no change in the total input. Under any other condition the input to the boiler whose curve has the greater slope is decreased more by a slight decrease in output than that of the other is increased by the compensating increase in output; so that the result is a net decrease in combined input. Additional progress in the same direction will result in further improvement until the distribution giving equal slopes is reached.

94 To determine graphically the condition for minimum combined input requires only a few simple "cut-and-try" steps. On the other hand, the analytical determination, while entirely possible, involves a highly undesirable degree of complexity.

95 It is the intention of the authors, by this short discussion, to make clear the practical justification of placing the limits specified at the beginning of Part II on the conditions; and also to point the way to such modifications in the method as are necessary in order to use the same principles in the many cases which cannot be so limited.

96 As a matter of fact, it will now be noticed that the second restriction mentioned is not an independent one, but a corollary of the first and the requirement of minimum combined input. If any number of steaming boilers are duplicates, their performance curves coincide at all points; consequently the points at which the individual curves have equal slopes coincide, the abscissas of the points are equal, and the plant load is equally shared among the boilers.

III—OPERATION OF BOILER PLANT AT MINIMUM TOTAL COST

97 Since formulas have already been developed in Part II giving the coal consumption in tons per day, the period for which the total cost will be determined is taken as one day.

98 The following additional symbols will be used in this

analysis, together with certain others which will be defined when they are introduced.

C_t = total cost of operating the boiler plant, dollars per day
C_f = total fuel expense, dollars per day
C_i = fixed charge on capital invested in boiler plant, dollars per day
C_s = boiler-plant operating cost less fuel, dollars per day (labor, supplies, etc.)
C_u = boiler-plant maintenance cost, dollars per day
k = ratio between the time required to clean and overhaul a boiler and the elapsed time from the beginning of one cleaning period to the beginning of the next
N_0 = number of boilers installed.

By definition,

$$C_t = C_f + C_i + C_s + C_u \quad . \; . \; . \; . \; . \; . \quad [\mathbf{33}]$$

and

$$\frac{dC_t}{dN_1} = \frac{dC_f}{dN_1} + \frac{dC_i}{dN_1} + \frac{dC_s}{dN_1} + \frac{dC_u}{dN_1}$$

99 Before this expression can be used in the determination of the value of N_1 resulting in minimum total daily boiler-plant cost, all of the components of the total cost must be expressed in terms of N_1.

Rationalization of Components of Total Daily Boiler-Room Cost

100 By the definition of k, when N_1 boilers are required at the time of the daily peak, kN_1 boilers must be undergoing overhauling, and

$$N_0 = N_1 + kN_1 = (1 + k)N_1 \quad . \; . \; . \; . \; . \quad [\mathbf{34}]$$

101 Careful analysis of the total fuel expense, C_f, shows that in most cases it is best represented by a formula such as the following:

$$C_f = V_2 + (V_3 + V_4 + V_5V_6)T_T$$

in which V_2 = cost of coal handling, in dollars per day, projected back to 0 tons handled per day

V_3 = cost of coal, F.O.B., in dollars per ton

V_4 = increment of coal-handling cost, in dollars, for each additional ton of coal handled

V_5 = cost of ash removal, in dollars per ton of ash

V_6 = percentage of ash as fired, expressed as a decimal.

102 This more exact expression may be replaced, to a very close approximation, for some considerable number of tons above or below a given initial value, by the shorter expression,

$$C_f = V_1T_T$$

since V_2, quite small as compared to C_f, is the only part of C_f not affected proportionately by a change in the number of tons of coal consumed.

$$\frac{dC_f}{dN_1} = \frac{dV_1T_T}{dN_1} = \frac{dV_1T_T}{dx_1}\frac{dx_1}{dN_1} = V_1\frac{dT_T}{dx_1}\frac{dx_1}{dN_1}$$

103 The value of $\frac{dT_T}{dx_1}$ has already been derived in the differentiation of Equation [**30**], while $x_1 = \frac{L_1}{H}N_1^{-1}$, and therefore $\frac{dx_1}{dN_1} = -\frac{L_1}{H}N_1^{-2}$.

104 The value of $\frac{dV_1T_T}{dN_1}$ may now be expressed, substituting for x_1 its value $\frac{L_1}{H}N_1^{-1}$, and multiplying the entire expression $\frac{dT_T}{dx_1}$ by $-\frac{L_1}{H}N_1^{-2}$:

$$-L_1^bH^{1-b}V_1K\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]a(b-1)N_1^{-b}$$
$$+HV_1K(c-0.202)\sum_{p=1}^{p=m}\left[h_p\right]+4.848HV_1K$$

105 The value of C_i is expressed very closely by an equation of the following form:

$$C_i = [S_1 + S_2N_0]r = [S_1 + S_2N_1(1+k)]r$$

in which S_1 is the investment (dollars) in the boiler plant, projected back to 0 boilers installed, S_2 the increment of investment (dollars) in the boiler plant for each additional boiler installed, and r the fixed charge rate per day against the boiler-plant investment.

$$\frac{dC_i}{dN_1} = 0 + rS_2(1+k)$$

106 The value of C_s is also expressed very closely by the linear formula

$$C_s = P_1 + P_2N_0 = P_1 + P_2(1+k)N_1$$

in which P_1 is the boiler-plant operating cost less fuel (labor, supplies, etc.) in dollars per day projected back to 0 boilers installed, and P_2 the increment of operating cost less fuel in dollars per day, for each additional boiler installed.

$$\frac{dC_s}{dN_1} = 0 + P_2(1+k)$$

107 Boiler-plant maintenance cost is one of the most difficult items of operating cost to correlate with the boiler operation which

has made the maintenance necessary; principally because in one year when a large output has been required of the boiler plant the maintenance has been allowed of necessity to pile up, while in a succeeding year of much lighter load boilers can be removed from service more easily and the maintenance which should have been performed the previous year can be attended to.

108 However, the analysis of such boiler-plant maintenance

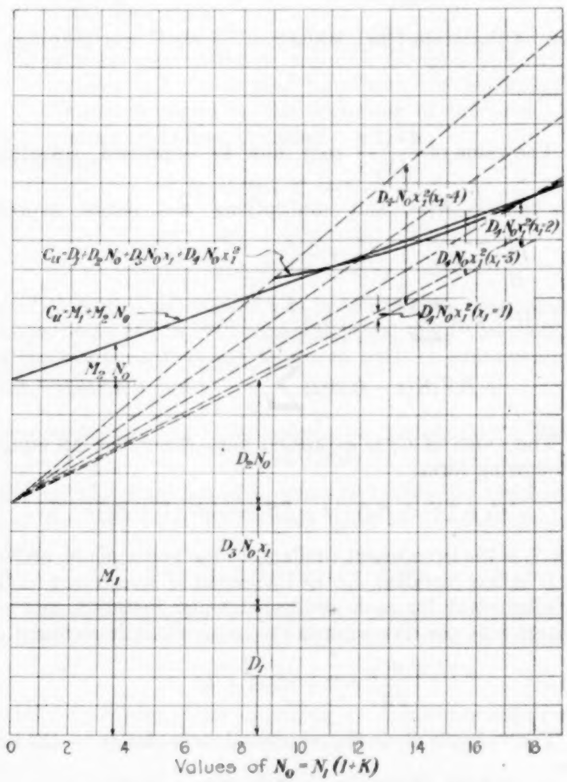

FIG. 21 GRAPH OF BOILER-PLANT MAINTENANCE COST ACCORDING TO EXACT AND SIMPLIFIED FORMULAS

data as have been available to the authors indicates that a formula to account satisfactorily for the variation of boiler-plant maintenance cost must contain at least (1) a constant term; (2) a term proportional only to the number of boilers installed; (3) a term proportional to the number of boilers installed and to the first power of the maximum output rate; and (4) a term proportional to the number of boilers installed and to a higher power of the maximum output rate than the first.

109 Expressing this analytically,

$$C_u = D_1 + D_2N_0 + D_3N_0x_1 + D_4N_0x_1^w$$

and, as $N_1x_1 = L_1/H$,

$$D_3N_1(1 + k)x_1 = D_3\frac{L_1}{H}(1 + k)$$

Since L_1 is held constant, this term is constant and is to be added to D_1.

110 Fig. 21 represents graphically the equation

$$C_u = \left[D_1 + D_3\frac{L_1}{H}(1 + k)\right] + D_2N_1(1 + k) + D_4N_1(1 + k)x_1^w$$

assuming $w = 2$. The locus of C_u is shown for a given maximum load, as N_0 varies. It will be noted that this locus is quite a flat curve throughout the range of practical maximum output rates. For the sake of simplicity and in view of the doubtful accuracy with which D_1, D_2, D_3, and D_4 can be determined, the curve has been replaced by the straight line

$$C_u = M_1 + M_2N_0 = M_1 + M_2N_1(1 + k)$$

assuming the daily boiler-plant maintenance cost to vary only with N_0. In this linear formula, M_1 = boiler-plant maintenance cost, in dollars per day, projected back to 0 boilers installed, while M_2 = increment of boiler-plant maintenance cost, in dollars per day, for each additional boiler installed.

$$\frac{dC_u}{dN_1} = 0 + M_2(1 + k)$$

111 Since all of the terms in the derivative of Equation [**33**] contain, as now expressed, only one variable, N_1, the solution of the derivative equation can be accomplished to determine the value of N_1 resulting in minimum total cost of operating the boiler plant per day.

$$\frac{dC_t}{dN_1} = \frac{dC_f}{dN_1} + \frac{dC_i}{dN_1} + \frac{dC_s}{dN_1} + \frac{dC_u}{dN_1}$$

$$= -L_1^bH^{1-b}V_1K\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]a(b-1)N_1^{-b}$$

$$+ HV_1K(c - 0.202)\sum_{p=1}^{p=m}\left[h_p\right] + 4.848HV_1K$$

$$+ rS_2(1 + k) + P_2(1 + k) + M_2(1 + k)$$

112 Equating to zero, substituting N_J for N_1 (where N_J is the value of N_1 resulting in minimum daily total cost of operating the boiler plant), and solving

$$N_J=\frac{L_1}{H}\left\{\frac{\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]a(b-1)}{(c-0.202)\sum_{p=1}^{p=m}\left[h_p\right]+4.848+\frac{(1+k)(rS_2+P_2+M_2)}{HV_1K}}\right\}^{\frac{1}{b}} \quad [\mathbf{35}]$$

113 In view of the complication and the number of parameters in Equation [**35**], no graphical solution of it has been provided. However, only two of the steps require anything but slide-rule multiplication and division, and arithmetical addition and subtraction. These two steps are (1) the determination of $\sum_{p=1}^{p=m}\left[h_p\left(\frac{L_p}{L_1}\right)^b\right]$ and (2) the application of the exponent $\frac{1}{b}$ to the quantity inside the large brackets.

114 The first of these already has a graphical solution provided in Fig. 18, used in the determination of R_2; the second is best accomplished by taking the logarithm of the value of the quantity inside the large brackets, dividing by b, and finding the antilogarithm of the quotient.

115 Usually, commercial load curves are very nearly alike for 250 to 300 days of the year. In some cases Saturday curves differ materially from those of week-days, while in other cases the Saturday and week-day curves are very similar. This means that the number of boilers indicated as the most economical to install by the week-day load curve should be given considerably greater weight in determining the actual number which will be most economical throughout the year than the number indicated by the Saturday or Sunday load curve.

DISCUSSION [1]

J. Harvey.[2] There is no doubt that the correct application of the formulas developed by Messrs. Funk and Ralston would provide a basis for the most economical boiler-plant operation possible. The following paragraphs constitute an attempt to evaluate the constants and determine the correct procedure for the Delray plant of The Detroit Edison Company.

This plant has two types and three sizes of boilers, with some variation in arrangement of baffles. Some of the boilers are equipped with economizers. Three different manufacturers furnished the stokers, all of the underfeed type. The sizes of the stokers are varied to suit the different sized boilers, and minor differences also exist among some of the stokers of the same size and make. The plant records do not include the data necessary to compute an accurate efficiency-output curve for each boiler,

[1] See also discussion by Leo Loeb, Paper No. 1911, p. 599.

[2] Boiler-Room Engineer, The Detroit Edison Co., Delray, Detroit, Mich.

nor indeed for any boiler. The most recent boiler test at the Delray plant which was sufficiently elaborate to permit the derivation of a reliable efficiency-output curve was made prior to the arrangement of the equipment in its present form. However, in 1921 boiler No. 14 at the Connors Creek plant was tested, and an accurate efficiency-output curve was derived. This curve was adapted to boiler No. 33 at Delray, which is similar to boiler No. 14 at Connors Creek. Boiler No. 33, under normal operating conditions, has a higher flue-gas temperature than was observed during the test of boiler No. 14, and a suitable correction was made to the efficiency-output curve.

Using this curve, it was possible to draw the input-output curve for boiler No. 33, and follow the procedure outlined by the authors for evaluating the constants in the equation

$$y = x + ax^b + c$$

To correct for steam consumption of boiler auxiliaries the results of the test of boiler No. 14 at Connors Creek were used. To correct for superheat, the superheat curve for boiler No. 27 at Delray and the guarantee conditions for turbo-generator No. 10 at Delray were used.

Using the above data the following values were found:— $a = 0.1127$; $b = 2.4$; and $c = 0.1873$. The fact that c was found a positive quantity is to some extent a check on the accuracy of the assumptions.

The computations from this point on were made on the assumption that all the boilers were identical with boiler No. 33, which, as shown above, is incorrect. R_1 was evaluated, and from it x_P, the output rate at which boilers should be banked. x_P was found to be 0.315, which is impractical and probably inaccurate.

Finally, using Equation [**12**], curves were drawn for Tons of Coal per Hour *vs.* Number of Boilers Steaming, for plant loads of 50,000 boiler hp., 30,000 boiler hp., and 15,000 boiler hp., respectively. These curves all indicate that the minimum coal consumption occurs at the minimum practicable rating on the boilers. It is well to remember, however, that the factor $d\epsilon^{-fx}$ has been neglected.

In connection with Equation [**12**] attention may be called to the fact that the authors use 6750 B.t.u. per boiler hp-hr. as the requirement for banking underfeed stokers, while from the test of Boiler No. 14 at Connors Creek, Mr. P. W. Thompson developed the equation for a "dead" bank:

$$\begin{aligned} \text{lb. coal} &= 8000 + 280\,x \\ \text{where } x &= \text{number of hours banked} \\ \text{boiler hp.} &= 2365.4 \\ \text{Average heating value of coal} &= 12{,}450 \text{ B.t.u. per lb.} \end{aligned}$$

For a banking period of 9 hours this equation gives 5550 B.t.u. per boiler hp-hr. Other writers in the current technical literature

give other values; it seems likely, therefore, that this figure varies for different installations.

No attempt was made to proceed further along the lines laid out by the authors. It was evident that a prerequisite to the application of these formulas to the operation of the Delray plant would be the derivation of at least nine input-output curves. This would necessitate an increase in the number and extent of the boiler-room records, and there is some doubt as to whether the expense of obtaining and keeping up to date of these records would be justified by the saving resulting from their use.

As to the feasibility of the authors' procedure, it appears that there are two ways in which the formula developed by them may be applied to the operation of a given plant:

1 To evaluate the constants on the assumption that the boiler plant is in first-class operating condition, and use the results obtained as a general guide to operation, and

2 To evaluate the constants on the basis of existing conditions, and operate as nearly as possible in conformity with the results obtained. This would necessitate immediate adjustment of the values to take account of such factors as the failure of part of a stoker, for instance, or the failure of a soot blower, or other similar factors.

The first of the two applications seems preferable. Presumably the boiler-room foreman would be a man of experience and judgment, and with these figures as a guide, he would be able to use the available boilers to the best advantage. The second alternative would entail continuous observations and computations, and doubtless in many cases would arrive at the same result as the first.

The Authors were much interested in the discussion by Mr. J. Harvey and desire to state that application 1 suggested by him is the one that the authors felt was by far the more logical. It is not the intent to apply the methods of reasoning outlined to everyday operation but to formulate the methods that should be used, and then operate as close to this as conditions permit, instead of operating in accordance with unguided judgment that might or might not be good.

Attention is called to Par. 58 where a specific note is made of the fact that if values less than 80 per cent rating are obtained, the effect of the term $d\epsilon^{-fx}$ must be considered. However, with the smaller quantity of coal used for banking the formula still points to the fact that the boilers should be operated at as low a value as physically possible before banking.

Referring to Mr. Loeb's discussion,[1] it is desirable to point out the fact that Mr. Loeb has used a very short range in boiler rating

[1] See Paper No. 1911, p. 599.

from which to make his deductions and therefore the conclusions that he draws are not general conclusions. Also, apparently Mr. Loeb has not carefully read the paper because he desires to know why the coefficient is unity instead of a decimal less than unity, that is, he asks why the input is fundamentally less than the output, giving an efficiency greater than 100 per cent. The reason that the coefficient of x is unity is stated in Par. 10 quite clearly.

Mr. Loeb's equation

$$Y = AX \div B$$

is apparently a linear equation. However, Mr. Loeb has neglected the fact that A and B are both variables with the output rate.

If it were true that the boiler input-output curve were a straight line, Equation [**4**] would still be as follows:

$$y = x + ax^b + c$$

in which b would be unity, and therefore the equation would reduce to

$$y = (1 + a)x + c$$

which is the linear equation that Mr. Loeb is seeking. This is merely an indication that the equation given by the authors is perfectly general and can be applied to any extreme shape of input-output curve.

The authors desire further to state that the form of this equation is not necessarily applicable to boilers alone but applies to all energy conversion equipment, either in the form given or in the more general form of Equation [**3**]. As the simplest example of this, the transformer may be taken, in which the iron losses are practically constant and the copper losses vary as the square of the load. The substitution of the iron losses for c, the copper losses for ax^b and the actual load transformed for x will give any transformer input-output curve with absolute accuracy. While this example may not be of interest to boiler operators, it is merely given as an instance of the generality of the form of the equation.

The authors are further at loss to understand why Mr. Loeb insists upon making a straight line out of his curves, implying thereby that it is possible to obtain output from the boiler without any input, which of course is impossible.

Mr. Loeb admits in his discussion that these curves must be turned to intercept the input axis rather than the output axis at zero load. In this Mr. Loeb's diagrams and discussion do not quite agree. If he had drawn his curves in accordance with his admission he would have secured curves similar to the authors' as inspection of Fig. 1, Curve B would indicate that between 1.0 and 2.0 a straight line could be drawn as readily as a curve. This is what Mr. Loeb has done.

No. 1913

POWER-PLANT ECONOMIES AS AFFECTED BY MODIFICATIONS OF THE HEAT CYCLE

INTRODUCTION

IN the past few years there has been considerable discussion of heat balance and heat-balance control as applied in detail to various station designs. In most cases, however, these stations have operated on the straight Rankine cycle.

The temperature range of this Rankine cycle as applied has been gradually increased, and although the knowledge has been available that various modifications of this cycle would considerably increase the overall efficiency, nevertheless, it was recognized there were many practical difficulties to be overcome. Several stations have been designed lately with marked deviations from the straight Rankine cycle.

It is timely that the possibility of these various deviations should be considered, both from an ideal point of view and as limited by practical and economic considerations.

The papers presented in the following discussion of the heat cycles call particular attention to the regenerative and reheat cycles, pointing out the possibilities both from a thermal efficiency and economical point of view, with a short discussion of the possibilities to be obtained by using combinations of the vapors from various liquids over a wide range of the heat cycle.

No. 1913 *a*

THE MARGINS OF POSSIBLE IMPROVEMENT IN THE CENTRAL-STATION STEAM PLANT[1]

By Ernest L. Robinson,[2] Schenectady, N. Y.
Member of the Society

The maximum attainable efficiency of the ideal heat engine is discussed, and the circumstances which limit it, in order to point out the various margins available for improvement along different lines. The arrangement of the circulating systems is considered and, after noting the ideal arrangement, practical layouts are taken up. The advantages of the mercury turbine and the steam-extraction cycle are emphasized, and the great increase in capacity rating which has been made possible by purely thermodynamic improvements is noted.

MANY influences are tending to force manufacturers of power to use the greatest care in the layout of their plants in order to obtain the best possible economy of operation. The greatly diversified inventions requiring the use of electricity have much increased the demand for it. Existing plants are more and more pressed to the limit of their capacities and must get the largest output from the available equipment. Increasing costs of coal continually tend to emphasize the importance of the fuel charge in the total costs of operation, and point to the necessity of greater economy in its use.

2 It is appropriate, therefore, to take stock of the processes and methods available for improving steam-plant efficiency. The engineering periodicals exhibit a great interest in these questions and a large variety of improvements have been suggested. Usually the authors have devoted themselves to the careful and ingenious application of one or several methods of improving efficiency and have worked out the practical application of their ideas in particular cases. The purpose of the present article is to run over the whole field in a very general way, and to point out the limiting efficiencies attainable and show the lines along which the greatest margins for

[1] For discussion and closure see pp. 766 and 802.
[2] Turbine Engineering Department, General Electric Company.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

possible improvement lie. At the outset, then, practical and mechanical limitations will be disregarded in order to go first to the theoretically best economy. Fortunately this may be done in plain engineering language without the use of complicated mathematics. When this has been done it will be appropriate to discuss briefly the various difficulties in the way of going the limit at once.

3 In the case of any particular installation the correct procedure would, of course, be to start with good practice and make such improvements as can be shown to be desirable. The object of the present paper will have been accomplished if, starting from the more remote ideal, it directs attention to the whole field open for selection and thus enables a more valuable choice to be made as to the best line of development to apply in a particular case.

The Ideal Conversion of Heat Energy

4 Fundamentally conceived, the steam plant is simply a heat engine for converting fuel into salable power. The textbooks outline certain theoretical facts about heat engines which it is convenient to recall at this time. A heat engine receives heat from a hot source, does work, and rejects heat to a cold receiver. The process of combustion may be considered as the hot source and the condenser as the cold receiver. Heat is directly convertible into other forms of energy at a fixed theoretical rate known as the mechanical equivalent of heat or the heat equivalent of electric energy. If the ideal heat engine could convert all of the heat into work at this rate it would be necessary to reject the working substance devoid of all heat content, that is, at absolute zero. A working substance to carry the heat is of course essential. If, at any time, energy is reclaimed from the working substance, without other heat exchange, its availability to do work has been decreased a corresponding amount, that is, its absolute temperature has been changed. In fact, this idea is the whole basis of the definition of the thermodynamic scale of temperature. Since the temperature of a condenser can hardly be lowered to absolute zero, it is necessary for heat engines in general to do work, even under ideal circumstances, at an efficiency far less than unity. This efficiency is represented by the actual temperature drop of the working substance divided by what the drop might be if all heat were abstracted, that is, the initial absolute temperature. This is the well-known Carnot-cycle efficiency and also the efficiency of any reversible engine.

5 As this idea of a reversible engine will come up again, it is well to recall it a little more carefully. It really must be the ideal engine because, if any other engine which might be thought more efficient — so that for a fixed amount of work done it has a smaller heat turnover — were used to drive the less efficient reversible engine as a heat pump, such a supposition would enable two

isolated machines working together to have a net turnover of heat from a cold to a hot source. Such a condition is contrary to all experience, and the reversible engine with the Carnot efficiency is thus vindicated as the ideal engine. Furthermore, the textbooks go on to show not only that the Carnot efficiency cannot be exceeded, but that any cycle including an irreversible process is necessarily less efficient. An illustration may help to recall this point.

6 The transfer of heat from a furnace at 2000 deg. fahr. to a boiler at 400 deg. fahr. is an irreversible process. Heat cannot be caused to flow by itself from the boiler to the hotter furnace. As a result of the decrease in temperature the availability of the heat energy in the steam at boiler temperature to do work is very much less than the theoretical availability of the same amount of heat energy in the furnace at the temperature of combustion. On the other hand, if the boiler absorbed the heat of the furnace at furnace temperature, such a process would be reversible, and would enable the steam to work with the higher thermodynamic efficiency which the high temperature would permit. In fact, the idea of availability to do work is synonymous with thermodynamic efficiency, and this idea of loss of availability or thermodynamic efficiency in connection with the irreversible step is the important thing to note at this time.

7 The Carnot cycle is made up of an isothermal expansion and an adiabatic expansion, followed by an isothermal compression and an adiabatic compression. The isothermal process or heat exchange without alteration of temperature can, of course, be worked either way since no temperature drop is involved. The adiabatic process is reversible because it simply means no heat exchange to the outside at all but simply a conversion of internal or heat energy into external or work energy. Ideally, the isothermal is infinitely slow and the adiabatic is correspondingly sudden in order to be reversible.

8 There is also the regenerative process in which all heat that passes out of the working substance during cooling is transferred at its own temperature to an auxiliary contraflow substance, so that it may be restored at its own temperature during a similar return process. This requires the infinite slowness of the isothermal and the supposition of a perfect contraflow heater always in contact with the working substance, so that the temperature differences between the working substance and the auxiliary substance are always infinitesimal.

Temperature Limits

9 Certain very elementary conclusions are possible at once from this review of simple thermodynamic principles. It is desirable to have as cold a receiver as possible, and since there are practical limits to this coldness, it is also desirable to have the

source of heat at as high a temperature as possible in order to give the widest possible operating range. These are the things that must be kept in mind when deciding on the efficiency of the ideal process.

10 With steam as a working substance and modern turbine construction it has been found that vacuums corresponding to temperatures between 70 deg. and 80 deg. fahr. are the best that can be obtained with the cooling water available. Of course, it is imaginable that during an Alaskan winter a turbine might be made to exhaust into an air-cooled condenser at a temperature well below 0 deg. fahr. Obviously in such a case the suitability of steam as a working substance would also have to be considered.

11 Likewise, at the other end of the process it happens to be far easier to produce high temperatures than to utilize their advantage, because, in the case of steam, the vaporization at a high temperature cannot be accomplished without confinement at an excessive pressure. Thus it is evident that the working temperature range is hedged around by various practical considerations which will need special consideration, and this will be given presently.

The Thermal Processes

12 It is next important to consider the thermal processes and how closely they may be made to correspond with the ideal arrangement, that is, what sort of cycles can be used to the best advantage. It has been pointed out that the thermal processes should be either isothermal, adiabatic, or regenerative, that is, in every case reversible. Beginning at the furnace, the fuel and air are brought together in the presence of heat. Combustion takes place and more heat is liberated. This is absorbed by the boiler and carried to the engine by the working substance. Finally such heat as has not been converted into work is rejected to the condenser. But in addition the burned gases escape up the stack.

The Ideal Furnace

13 If, for the present, attention is confined to the furnace, it is plain that all the substance, fuel and air, which goes on to the grate and is heated must, in exactly the same quantity, disappear up the stack. Furthermore these substances which start at atmospheric temperature rise to the furnace temperature and then are returned to the atmosphere, eventually cooling to their initial condition. Considering the approach to and exit from the furnace, it is plain that equal quantities of substance rise on the one hand and fall on the other through equal ranges of temperature. This suggests the recovery of all the waste heat of the flue gases by regenerative air preheaters. Ideally, these should be contraflow, with such surfaces and velocities that all the heat of the burned gases is transferred to the incoming air — and fuel also.

14 Next, considering the process of combustion, it may be regarded as a simple liberation of heat which occurs when the fuel and air are brought into contact at a high temperature. Suppose the fuel to be insulated from the air during preheating until the furnace temperature is reached, when combustion is permitted. Suppose the boiler to be so large that it will absorb the heat of combustion immediately without allowing any appreciable difference of temperature, the working substance being, in the ideal case, at the furnace temperature. This amounts to an isothermal transfer of heat and completes a reversible cycle in the furnace, at least from a thermodynamic point of view. The chemistry of combustion is another question. Still retaining the purely thermodynamic viewpoint, the furnace temperature is ideally arbitrary.

15 Furthermore, if the boiler is operated at a lower temperature than the furnace, the consequent loss of availability, according to the Carnot efficiency, must be charged against the boiler or its working substance and not against the furnace, since all the heat of combustion would still be transferred to the working substance. On the supposition of a perfect transfer of all the heat of the flue gases to the entering air and fuel, there would be no thermodynamic loss in the furnace, and if, in addition, an isothermal transfer of the heat of combustion to the boiler could occur, then the boiler could be included in the statement.

The Engine. Rankine Cycle not Ideal

16 Turning now to the working substance in the engine, the steam is the first consideration. It is not necessary to discuss the Rankine cycle for steam. Instead of the adiabatic compression of the Carnot cycle an increase of pressure at constant volume is used, a process which is not reversible. The thermodynamic efficiency has been calculated from the steam tables and plotted in Fig. 1 for dry saturated steam and for various initial temperatures and pressures. A glance at the diagram shows that the higher the temperature, that is, the higher the ideal efficiency, the farther does the Rankine cycle fall short of it.

17 This is especially true in the case of superheat, even though it improves the actual efficiency. The ideal efficiency rises enormously faster than the change in Rankine efficiency with superheat. With saturated steam the conversion from liquid to steam in the boiler, the expansion in the turbine, and the condensation are all reversible processes, only the process of heating the liquid being irreversible. On the other hand, with superheated steam the absorption of heat is no longer isothermal, so that in this case two out of the four processes which go to make up the cycle are irreversible. Hence it is reasonable that the latter cycle should fall farther below the ideal. Moreover it follows that for any particular temperature the less the superheat the higher will be the efficiency. In fact, for a particular temperature the efficiency is

far the best if the pressure is high enough for the steam to be saturated.

The Extraction Cycle

18 The question now arises as to what cycle can be used in order to cause the steam to turn over the heat in accordance with the best theoretical standards. Obviously, the superheated-steam cycle is far from ideal. In fact, it is hard to justify the use of superheat from a purely theoretical viewpoint, although its application is thoroughly justified by a variety of practical reasons which will be discussed eventually. The Rankine cycle with saturated steam has only one process, the feed heating, which is not reversible. The idea of a regenerative process consists in many small exchanges of heat occurring at successively different temperatures in infinitesimal steps. Suppose the steam-extraction process be pushed to such a limit. It truly has been called a regenerative process.

19 Suppose steam is extracted from every stage of the turbine and that the number of stages is indefinitely increased. If at each stage just enough steam is extracted to heat the feed the infinitesimal temperature difference, each bit of steam so extracted has gone through a cycle reversible except for the infinitely small temperature drops during the feed heating, and these infinitesimal temperature drops occur in an unlimited number of small steps as in any regenerative process. The boiler temperature can thus be reached within an infinitely small amount. In other words, the extraction cycle as a whole is entirely reversible in the limit.

20 In order to understand clearly the truth of this statement, the reversed cycle is traced as follows: Starting with a hot liquid, it is caused by a very small cooling to evaporate a part of itself isothermally at a reduced pressure and that part is compressed adiabatically to the higher pressure and temperature in the boiler where it gives up its heat isothermally and condenses into the hot liquid. Further cooling will evaporate a further quantity at a still lower pressure, and so on. If done in a very large number of steps the process may be rendered as close as desired to truly reversible.

21 The steam-extraction cycle with saturated steam is thus, in the limit, a truly reversible process with the ideal efficiency of the Carnot cycle. Furthermore, the demonstration has assumed only the pressure-temperature relation of a wet vapor. It is good not only for steam, but equally correct for saturated ammonia or saturated mercury. However, it does not apply to the case of a superheated vapor. This is an important fact to understand clearly. The equivalence of the Carnot and extraction cycles was demonstrated in the days of the steam engine before the common use of superheat, and as a result the literature on the subject is not clear that the inference is true only for wet vapors and not for superheat. Furthermore, it is easy to demonstrate

this equivalence analytically on the basis of certain assumed algebraic relations of properties. Such demonstrations, using algebraic relations which are easily shown to be inexact, have failed to command confidence in the conclusion that the extraction cycle really is equal, with an unlimited number of heaters, to the Carnot cycle.

22 This is about as far as abstract theory alone can go for the steam plant. The furnace and boiler have been imagined ideal

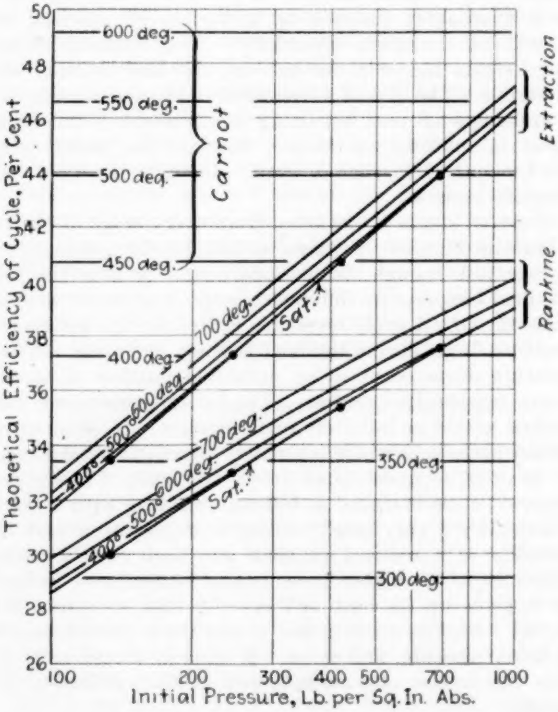

FIG. 1 THEORETICAL EFFICIENCIES OF STEAM CYCLES

(Based on Goodenough, 1917. Curves are for various initial total temperature deg. fahr., as indicated. Back pressure, 1 in. Hg)

by the use of regenerative air preheating with the flue gases, and the turbine has been imagined ideal by using saturated steam and heating the feed to boiler temperature by extraction. Thus the total margins for improvement are shown very clearly in Fig. 1, as explained before. The efficiency of the extraction cycle including superheat has been calculated according to the method outlined in the Appendix. It now becomes necessary to narrow the ideal still further by taking up the practical matters which deter-

mine the temperature limits and see what are the relative margins to gain among the various methods of increasing the efficiency. There are two principal questions: First, the temperature limits, and second, for established limits, how to approach as nearly as possible the ideal cycle.

The Upper Temperature Limit

23 The upper temperature limit is determined by the materials of construction rather than by the furnace or the working substance. Turbine materials maintain their strength, roughly speaking, up to 700 deg. fahr. Turbine steels are usually good up to 800 deg. fahr., but the gradient of decrease in strength per degree rise of temperature becomes so steep at higher temperatures that only very low stresses are allowable. Conservative practice therefore places 750 deg. fahr. as the limit at the present time. It has been made clear that, unless saturated steam is used, the efficiency of the process will fall far below the ideal. And with saturated steam these temperatures are above the critical and the pressure above 3000 lb. per sq. in. Both the excessive pressure and excessive moisture during expansion make it undesirable to use saturated steam at such temperatures. The curves for the extraction cycle and the Rankine cycle in Fig. 1 show very plainly the magnitude of this restriction due to the pressure of steam at saturation temperatures. The horizontal lines indicate Carnot efficiencies, and it should be noted that extraction efficiencies and Carnot efficiencies for particular temperatures become equal at the saturation line.

24 The temperature-entropy diagram is useful for visualizing the differences in the processes. Fig. 2 shows the Rankine cycle *ABCD*. The heat put in is represented by the area *FABCDE* and the heat exhausted by the area *ADEF*. The Carnot cycle working between the same temperature limits has an efficiency which might be represented by certain areas constructed by drawing the lines *AG* and *GB*. For the same heat exhausted as in the Rankine cycle, the Carnot cycle would have to absorb heat *FGCE* and deliver work *AGCD*. It is plain, therefore, since the extraction cycle is equal in efficiency to the Carnot cycle, that it is represented by these areas, although only as areas, because the broken lines do not represent properties of the substance. Now with the superheated Rankine cycle the work area *DCHK* is added out of a heat addition of *ECHM*. The efficiency is raised because the added work area is a larger ratio of the added heat area than the original efficiency. But the work area of the Carnot cycle for the elevated temperature is represented by the area *APHK*, so that the small addition *DCHK* is a ridiculously unsatisfactory approach.. Moreover the constant pressure line *BCH* is no longer isothermal. It is still possible to extract steam and heat the feedwater to the boiler temperature so that the efficiency

may be made equal to and even slightly better than the Carnot efficiency for the saturation temperature *GBC*, but the feedwater cannot be heated to the upper temperature limit without encountering excessive pressures. Hence this large area between the saturation temperature and the upper temperature accounts for the failure of the superheat cycle to meet the requirements of an efficiency corresponding to its temperature.

The Advantage of High Pressure

25 It should now be clear why there is a theoretical advantage in the use of high pressures, even though the temperature after

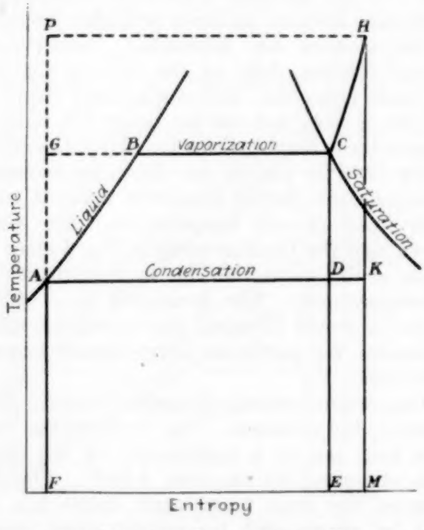

Fig 2 Temperature-Entropy Diagram for Rankine Cycle with Superheating

superheating is already at the limit fixed by the materials. In order to get the thermodynamic advantage of the highest temperature reached the steam must be generated at that temperature, a process which is practicable only for wet steam and provided that the required pressure is maintained. Thus it is clear that, due to the nature of a vapor, the pressure is quite as important as the temperature. The curves of Fig. 1 set forth the very material rise in theoretical efficiency with increase of pressure at various initial total temperatures, and it should be noted that the increase is still more rapid when steam extraction is employed. These curves have been carried to 1000 lb. per sq. in. absolute pressure. The manner of plotting on a logarithmic base renders the line so nearly straight that extrapolation is easy although it may not be war-

ranted. Even at 1000 lb. per sq. in. the properties of steam are not known with precision. The pressure-temperature relation is, however, well defined and this shows that with saturated steam at 1000 lb. per sq. in. absolute pressure, using the extraction cycle, the efficiency, high as it is, only equals the Carnot efficiency for

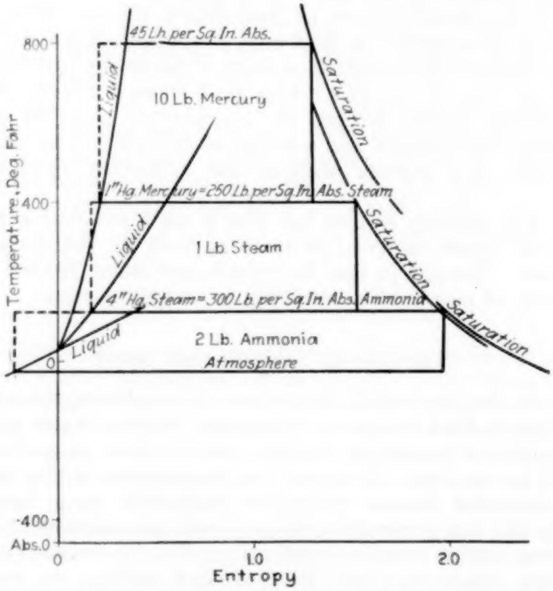

Fig. 3 Temperature-Entropy Diagram for a Combination of Mercury, Steam, and Ammonia

542 deg. fahr. and the temperature at the critical pressure itself is only about 700 deg. fahr.

The Binary-Vapor Turbine

26 The foregoing all points to the necessity of a substance whose vapor pressure will not be excessive at the temperature to which the materials of construction may be submitted. And this is the great advantage of the mercury turbine. Fig. 3 presents a temperature-entropy diagram for one pound of steam with a similar diagram for ten pounds of mercury drawn at the higher temperatures. In a particular machine the relative amounts of steam and mercury would be slightly different, depending on the temperature of the condenser-boiler. The ten-to-one ratio is convenient for plotting. The low pressure of 45 lb. per sq. in. at 800 deg. fahr. is very satisfactory and the exhaust at one inch of mercury back pressure heats a steam boiler at 400 deg. fahr.

and 250 lb. per sq. in. pressure, the steam from which may in turn be expanded to one inch of mercury back pressure. The entire working range of temperatures is from 800 deg. to 80 deg. fahr., and in each case wet vapors are used. In each case, also, extraction may, imaginably at least, be employed, although the mercury has so little liquid heat, as a glance at Fig. 3 shows, that its Rankine cycle is considerably nearer to ideal than is the case with steam. However, considering the ideal case, a simple scaling from Fig. 1 shows a theoretical increase of economy of the process just described of 37 per cent, as compared with good modern turbine practice using steam at 350 lb. per sq. in. pressure and 700 degrees fahrenheit temperature, and a saving of 28.7 per cent as compared with the same turbine utilizing the extraction cycle to the limit.

27 The mercury turbine has thus given a practicable way to reach the upper temperature limit as fixed by the strength of materials. The aim in that direction is now being directed at the materials of construction, and this is entirely correct.

The Lower Temperature Limit

28 At the other end of the process, the condenser, the temperature limit is fixed by natural conditions. Cooling water sufficient to maintain a vacuum at 80 deg. fahr. is about as good as can usually be expected. However, it is the intention of this paper to look somewhat beyond immediate possibilities in a variety of directions. Fig. 3 also includes the liquid and saturation lines for ammonia, and if a turbine were supposed to be equipped with an ammonia condenser-boiler, then another turbine, an ammonia turbine, might as easily continue the process as the steam turbine continued the mercury process. If such a steam plant were located in the polar regions in winter so that the atmosphere could condense the ammonia at – 25 deg. fahr., it would be possible to expand the ammonia to atmospheric pressure. Now if the feed is heated by extraction as in the case of the steam and mercury, the efficiency would be such as to effect a saving of 45 per cent over the case of a steam turbine working with the Rankine cycle at 350 lb. per sq. in. pressure and 700 degrees fahrenheit temperature.

29 It may also be noted here that sulphur dioxide could be used with less severe pressures than ammonia. This might be a matter for consideration if the process were adopted, and there is another reason than the efficiency of the process for the use of a refrigerating fluid for the last few stages. The capacity of the compound unit, being dependent on the last wheel of the steam turbine, could be very much increased if the steam expansion were stopped at a moderate vacuum and the remaining energy utilized in one of the suggested refrigerating fluids which have very much smaller specific volumes at the temperatures in question.

The Advantage of Steam Extraction

30 Having thus discussed what the theoretical efficiency is and how various practical limitations determine what is ideal in any particular case, the next matter to consider is the utilitarian question of the value of the various improvements already instituted. The idea is to take up step by step various limitations that lie between the application of the perfectly general theory and present steam-plant methods. The advantage of the mercury turbine has been pointed out. The theoretical advantage of steam extraction is shown in Fig. 1. The practical advantage depends on the number of heaters. Very roughly it may be said that, in comparison with no feed heating, the use of one heater will realize between a third and a half of the full theoretical gain, and also, very roughly, that each additional heater will realize an improvement about half as large as the last preceding heater added to the system.

The Advantages of Superheat

31 The advantage of superheat, as mentioned before, is almost entirely a matter of practical application due to the limitation of a high initial pressure. The desired temperature cannot be attained with wet steam without incurring too high an initial pressure. Hence the pressure is pushed as high as possible and the slight advantage illustrated in Fig. 2 is realized by superheating. The theoretical value is plotted in Fig. 1 in terms of total temperatures. Fortunately, the use of superheat is attended with a non-thermodynamic improvement in efficiency which is roughly of the same order of magnitude as the theoretical improvement. This is attributed to removal of moisture in the lower stages of the turbine. And still further the greater heat turnover per pound of steam results in an even greater reduction of water rate, which is advantageous in permitting a larger capacity rating of the machine.

32 In order to visualize these advantages it may be said, very roughly, that the theoretical gain in economy is about 1.5 per cent for 100 deg. fahr. superheat, while the non-thermodynamic gain is about as much more, making an overall reduction of heat rate in the neighborhood of 3 per cent per 100 deg. of superheat. The reduction in water rate is, on the other hand, of the order of 8 per cent per 100 deg. fahr. However, this must be clearly distinguished from an increase in the economy of operation of the plant. A reduction of water rate does not mean a corresponding reduction in the fuel rate because reductions in water rate usually entail a substantial increase in the heat per pound absorbed by the steam in the boiler.

Resuperheating

33 The desirability of resuperheating is due largely to the same reasons that apply in the case of initial superheat. But the

newness of the process makes it less easy to say what the gain may be beyond the theoretical amount. Fig. 4 shows at a glance that the cycle has been improved, roughly, by nearly the same degree as in the case of initial superheat as compared with saturated steam. The process is accompanied by a further reduction of water rate, as explained before, which is greater than corresponds with the reduction in heat rate. In passing it may be noted that this resuperheating process is a sort of partial expansion under isothermal conditions. For instance, if the resuperheating should

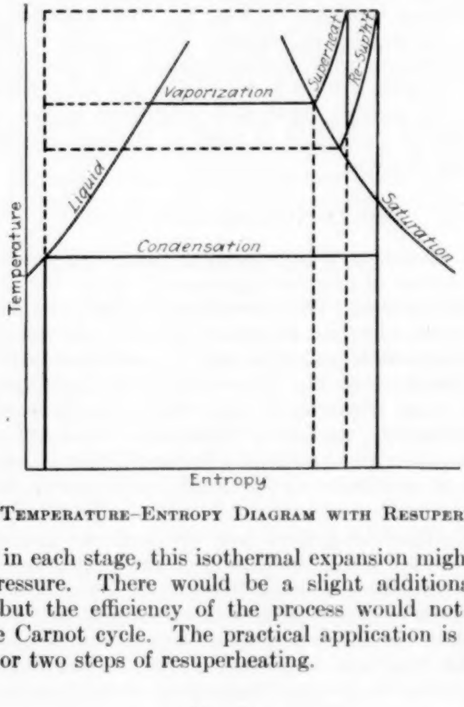

Fig. 4 Temperature-Entropy Diagram with Resuperheating

take place in each stage, this isothermal expansion might go on to exhaust pressure. There would be a slight additional gain in economy, but the efficiency of the process would not approach that of the Carnot cycle. The practical application is limited to about one or two steps of resuperheating.

Drying

34 Drying may be accomplished by the same process as resuperheating, namely, by the addition of heat; but if the addition of heat is only sufficient to dry the steam without superheating it, the gain will be slight and due largely to increased mechanical efficiency in the process of energy conversion. On the other hand, if the drying is done by moisture abstraction, the gain is very appreciable. The moisture removed should be used for feedwater heating, and the practical rise of efficiency due to drier steam in the turbine should be realized. Here again it is impossible to say how much improvement can be realized until the method is put

into practice. On the other hand, steam separators have been used for a long time, and the use of some form of separator in connection with extraction for feed heating would enable the removal of an especially wet sample of steam. In other words, instead of extracting a homogeneous sample, the extraction of all the moisture and only so much dry steam as might be required should enable one extraction drier used for feed heating to accomplish about as much gain in economy as, say, two extraction heaters of the surface-condenser type. This rough statement depends for its truth on the possibility of satisfactorily operating the drying process.

The Economizer

35 In all that has been said so far, there has appeared no place for an economizer. Indeed, as far as the theory goes, it appears to be anomalous. In considering the ideal furnace it was pointed out that flue-gas heat should be transferred to the entering air, and in considering the extraction cycle the logical source of feed-water heat was seen to be extracted steam. The economizer is intentionally aimed to extend the boiler process to a temperature below that corresponding to its pressure; i.e., to destroy its isothermal nature. But in all these cases where a substantial practice has grown up there is always a good common-sense reason sufficient to justify it at the time. Whether the practice should continue or not is always an appropriate question. The discharge of hot flue gas was a waste. To pump cold water into a hot boiler was bad. The transfer of heat from hot gas to water was a process well known, since it was taking place in every boiler; hence the economizer.

36 To secure an exchange of heat between large quantities of air with reasonable temperature differences is not so easy. The design of such apparatus is progressing at the present time, however, and installations of this nature have been made. It is certainly a process theoretically of such distinct value as to warrant serious attempts to accomplish its practical use. On the other hand, the success of this process will not in itself displace the economizer. The economizer works at temperatures roughly between 300 and 600 deg. fahr., whereas grate manufacturers prefer the air to enter the furnace at a temperature less than 300 deg. fahr. or certainly not much more. This is another very real limitation on the realization of the ideal arrangement. The inability to utilize this flow of fluid for returning low-grade heat to the furnace means definitely a corresponding amount of waste. Here is a very good reason for pulverized fuel, if it will permit the use of very hot air going to the fire.

Air Preheating from the Economizer

37 There are several alternative methods of connecting up these channels for the return of low-grade heat to the system. In

examining them it should be kept in mind that the idea is always to transfer heat with the least possible temperature difference. That is, the process should be as nearly truly regenerative as possible.

38 One suggestion that has been made with a view to transferring the heat of the exhaust gases to the incoming air is to fill the economizer with an auxiliary hot-water heating system and pipe this to hot-water radiators in the inlet-air system. Such an arrangement would avoid the bulkiness of air-to-air heaters. But it does not commend itself on account of its complexity, although the contraflow feature of a regenerative process might be utilized at each end. Moreover, the water would have to be confined under a very considerable pressure in order to attain the necessary temperature. Such a system would, however, leave the feedwater free to absorb the heat of extracted steam, and perhaps a more suitable auxiliary liquid than water could be selected for the transfer of the heat of the flue gases to the air.

Air Preheating with Extracted Steam

39 The use of extracted steam for air preheating is another scheme. If the economizer is used at the same time, it will be observed that the recommended arrangements of the ideal process have been crossed. In detail, instead of recovering the low-grade heat by the two regenerative processes, flue gas to air and extracted steam to feedwater, the following processes have been substituted: Flue gas to feedwater and extracted steam to air. This is all right as far as it goes, but examination shows that the crisscross is not equivalent to the direct process. To make this plain it is noted that very roughly two pounds of air flow are required to one pound of water, whereas the specific heat of the air is only about one quarter as much as that of the water. The air circulation, then, can carry back only about half as much heat per degree rise as the water circulation. There is also the limitation of grate temperature. Thus, even if all the air should be preheated by extracted steam, it would still be desirable to extract more for feed heating.

Parallel Feed Heating

40 Still another arrangement occurs to the author. The economizer must receive reasonably cold water in order to work efficiently, whereas the extraction process is not pushed to its limit unless it heats the water to boiler temperature. Present arrangements send the feedwater from the heaters to the economizer, thus dividing the available temperature range between the two heating devices. A more efficient way than to divide the temperature rise would be to divide the circulation into two very roughly equal parts (supposing pulverized fuel and no limitation on the air-preheating temperature). One branch of the system

would be sent to the economizer as it comes from the hotwell and should permit the cooling of the flue gases to a materially lower temperature than with feedwater already preheated. The other branch of the system should be arranged to receive extracted steam, including appropriate amounts from the high-pressure stages, since these temperatures no longer detract from the action of the economizer. Similar amounts of extracted steam would be used for preheating the air. Such a process in perfect adjustment would approach the ideal process.

The House Turbine

41 It is now time to discuss the use of a house turbine for feedwater heating. Very simply, if the house turbine has the same efficiency as the main units it is exactly equivalent to single-stage extraction. A house turbine is rarely as efficient as the main units, and generally means a less economical heat rate than extraction from the main units. On the other hand, it has certain advantages in the way of flexible adjustment that cause its use as a so-called "heat-balancer" element. By shifting the load on the house turbine the temperature of the feedwater may be regulated. It would be preferable to control the quantities extracted from the main units.

Steam Auxiliaries for Feed Heating

42 If steam auxiliaries are used for other reasons, the exhaust heat from these may be used for feedwater heating. It is interesting to read descriptions of this common practice which seem to reflect an opinion that it is desirable to create a number of losses in order to have losses to recover. Steam auxiliaries are usually very inefficient as compared with the main unit and therefore not to be compared with motor drive. In consequence the use of such heat is a waste as compared with extraction from the main unit. Furthermore, this heat is rarely added regeneratively but is dumped in at once, so that it reduces the ability to extract other steam more than in proportion to its own value.

43 The wisdom of having certain auxiliary drives duplicated by steam stand-bys in the interest of station reliability is not disputed at all. But it is maintained that the deliberate layout of a plant to heat the feedwater with such steam on the supposition that it is not lost is wrong, because such a layout prevents the use of another arrangement which is more efficient. The heat returned to the system is not lost, but for a given quantity of heat recirculated the auxiliaries can turn less into useful work than the main units.

Other Sources of Low-Grade Heat

44 There are also several other sources of low-grade heat about the station which may be used for feed-heating purposes. Among

these may be listed the following possibilities: The exhaust from the turbine high-pressure packings; the heat of the steam air ejectors; the heat of the bearings as recovered from the oil coolers; the generator losses as recovered in the generator cooling-air circulation. In such cases it is perfectly plain that the loss should not be created in order to get the heat. But having the heat it is allowable to use it, especially if it does not hinder the use of other heat. The process always should be as nearly regenerative as possible, and the final test of usefulness is the effect on the heat rate of the station as a whole as compared to the effect of some other arrangement.

The Final Criterion

45 The importance of considering the station heat rate as a whole is not always appreciated. As soon as the boiler plant and the turbine plant are connected by the regenerative processes the whole station becomes a single thermodynamic unit, and it is no longer possible to judge its economy by considering separately the boiler and the turbine. For example, the economizer may be made extremely efficient at the expense of all feed heating by extraction. Or the reverse may happen so that there is excessive waste in the flue gases. The only safe rule is to look at the heat rate of the whole plant. Furthermore, in accepting such a criterion the electrical apparatus must not be lost sight of, nor the fixed charges. No one would consider for a moment doubling the investment to obtain a one per cent increase in economy, and it is unwisdom of the same sort that focuses attention on any one element of the plant to the exclusion of its effect on the whole.

Capacity Rating

46 In conclusion, it seems necessary to consider the effect of the various improvements on the capacity of a unit. This relation has been noted repeatedly in passing, and it is a fortunate circumstance that improvements in efficiency usually result in a greater power output from a given amount of working substance and thus tend to increase the rated capacity of a machine. In the case of a particular turbine of large size, the rating depends on the volume of steam passing the last wheel so that reductions of water rate, especially water rate at the last wheel, cause corresponding increases in the rated capacity of the turbine. The reduction of water rate with superheat has been mentioned. A similar reduction occurs with increased pressure. The effect of steam extraction is obviously to reduce the water rate at the last wheel. Recent advances in the rating of single turbine units have brought the importance of this subject to the front, and it seems necessary to point out here the great increase in capacity which theory alone provides in the case of machines using the improvements which have been discussed.

47 Fig. 5 is a chart of reciprocal water rates or theoretical capacities in kilowatt-hours per pound of steam at the condenser. Curves have been plotted for the Rankine cycle with superheat and for the extraction cycle. Although the specific volume at the exhaust does not vary greatly, very fine lines have been added to show its theoretical value. A comparison with Fig. 1 shows at a glance how much more rapidly the capacity rises than the effi-

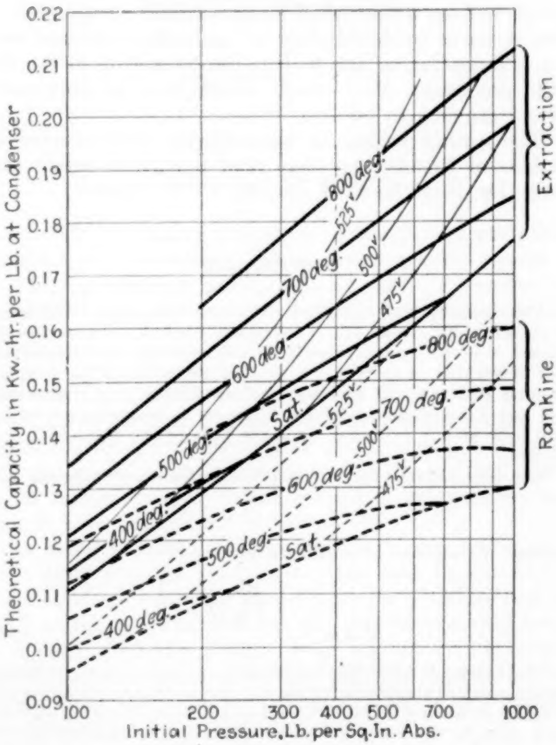

FIG. 5 THEORETICAL CAPACITIES OF STEAM CYCLES

(Curves are for various initial total temperatures in deg. fahr., as indicated. Back pressure, 1 in. Hg. Fine lines show exhaust specific volumes in cu. ft. per lb. Based on Goodenough, 1917.)

ciency. For instance from 130 lb. per sq. in. absolute and saturated steam with the Rankine cycle to 750 lb. per sq. in. and 750 deg. fahr. temperature with the extraction cycle, the capacity is just doubled while the economy is improved one third. In this example the theoretical specific volume at the exhaust is the same in each case.

48 In the case of increased superheat at a fixed pressure the

capacity rating of the machine increases less rapidly than the capacity per pound on account of the greater exhaust volume. On the other hand, with increased pressure at a constant superheat the capacity rating increases more rapidly than the capacity per pound because of the reduced exhaust volume. Generally speaking, increases of superheat have been accompanied by increases of pressure, so that exhaust specific volumes have been changed very little. It should therefore be plain that the continual increases of turbine rating which have occurred in the past decade without recourse to double-flow or multi-flow construction have been in a large degree due to true improvements in the thermodynamic processes. How much farther such improvements can be carried remains to be seen.

49 The author wishes to acknowledge the valuable critical suggestions of friends who have read the paper before printing, especially Dr. B. L. Newkirk and Mr. G. B. Warren.

APPENDIX

50 The formulas for calculating the theoretical value of steam extraction as shown in Fig. 1 and Fig. 5 are given herewith. One pound of steam is supposed to go to the condenser. At any pressure during expansion the quantity of working steam is $(1 + w)$ lb. At this pressure the corresponding total heat is H and liquid heat is h. The subscript 1 indicates the initial pressure and 2 the exhaust pressure. The formulas are expressed in terms of liquid heat at any pressure. e is the base of the Napierian system of logarithms.

51 The heat balance for any particular infinitesimal heater gives the relation:

$$dw(H - h) = (1 + w)dh$$

Integrating, the amount of working substance is:

$$1 + w = e^{\left[\int_{h_2}^{h} \frac{dh}{H - h}\right]}$$

52 The energy E added to the Rankine cycle per pound of steam in the condenser, which is also the extra amount put in by the boiler per pound of steam in the condenser, is given by integrating the following expression for the work done by the steam going to any particular infinitesimal heater.

$$dE = (H_1 - H)dw = \frac{H_1 - H}{H - h}(1 + w)dh$$

$$E = \int_{h_2}^{h_1} \frac{H_1 - H}{H - h}(1 + w)dh$$

53 These formulas are used by tabulating or plotting the quantities as functions of the liquid heat. The integration is then performed graphically and the efficiencies and capacities are easily calculated.

No. 1913 *b*

HIGH PRESSURE, REHEATING, AND REGENERATING FOR STEAM POWER PLANTS[1]

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and

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Members of the Society

In this paper the authors discuss the relative thermal and investment costs involved in modern turbo-generator stations, using various steam pressures and temperatures, and six different cycles. Particulars regarding the study are summarized in the three paragraphs immediately following.

THIS paper is a study of the relative thermal and investment costs involved in a modern turbo-generator central station, as determined by a consideration of steam pressures from 200 to 1200 lb. per sq. in., steam temperatures of 700 and 800 deg. fahr. and six different cycles in which reheating and regenerating are involved in various degrees.

2 The work naturally divides itself into two parts, the first of which deals with the information that may be gained from a study of the ideal cycles, the second with the probable fuel economies and operating characteristics that may be expected from real plants operating on the several cycles and at various pressures. The second portion of the paper also includes the relative investment costs which have been worked out in so far as they are affected by the various cycles and steam pressures.

3 In both parts special effort has been made to keep in mind the importance of correct relative, rather than absolute, values. In the first part the properties of high-pressure steam, which are uncertain to some degree, affect somewhat the absolute values of the efficiencies, but if the correct thermodynamic relations have been used the relative values cannot be very far from true. In

[1] For discussion and closure see pp. 766 and 802.

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the second part, however, there are so many complex relations involved that even the relative thermal values are necessarily somewhat uncertain. The estimated costs of the high-pressure stations operating on different cycles are based on information which is the best obtainable by the authors at this time, but naturally these estimates may be rather far from what will develop when such plants are more carefully studied or actually built.

I — IDEAL STEAM-POWER-PLANT CYCLES

Meaning of Ideal Cycle

4 An ideal steam-power-plant cycle is one in which water and steam are assumed to pass through a series of physical processes in a perfect fashion so that the losses due to throttling, turbulence, friction, radiation, conduction, and other phenomena unavoidably present in real plants, are excluded. The only loss that remains in such a case is the thermodynamically unavoidable one due to the heat rejected in the exhaust.

5 The actual steam plant, unfortunately, cannot be operated under such ideal conditions, but nevertheless the study of ideal cycles is of particular importance to engineers at the present time because they are working in the midst of many new developments relating to very high pressures and various schemes for reducing the exhaust losses. The exhaust loss in a real plant is larger than all the others combined, so that the best way of utilizing some of this exhaust heat, other than by industrial heating — which is sometimes feasible — becomes a problem of increasing importance as higher fuel prices are encountered.

Temperatures Considered

6 In this study it is assumed that at the present time the upper limit of the temperature of steam that can be used successfully in steam power plants is from 700 to 725 deg. fahr. The ideal-cycle diagrams have therefore been drawn with solid lines for a temperature of 700 deg. and with dotted lines for a temperature of 800 deg., a value which may possibly prove feasible in the next few years. The ideal-cycle calculations have been made for both temperatures, and the corresponding efficiency curves have been drawn solid and dotted, respectively.

Names and Diagrams of the Cycles

7 In considering the ideal cycles for any type of heat engines, that of Carnot is always entitled to first place. This cycle is too well known and too difficult to approach in a real steam plant, to require at this time any discussion or diagram. Attention will simply be called to the fact that with the upper steam temperatures of 700 and 800 deg. fahr. and an exhaust temperature of 79 deg., the corresponding Carnot efficiencies are respectively 0.535 and

0.572, which are, as all engineers know, the maximum attainable efficiencies with these temperature limits. Of the six other cycles considered, the Rankine is the only one that is almost universally called by the same name throughout the engineering world. In order to avoid confusion it will therefore be necessary to assign names to each of the cycles considered. As a matter of additional convenience for this study, the different cycles have been designated respectively by the first six letters of the alphabet. The names chosen are intended to be characteristic of the most unusual parts of the cycle, or in other words to identify the cycle

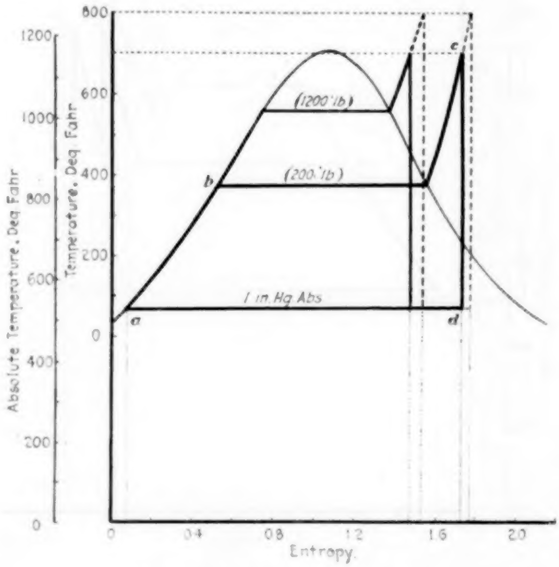

Fig. 1 Cycle A (Rankine)

For throttle pressure of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

by a name which indicates the unusual processes involved. All of the cycles herein considered have certain common characteristics, namely, the constant-pressure processes represented by the lines *abc* in each figure.

8 *The Rankine Cycle* (A) is made up of the processes indicated in Fig. 1, in which the cycle is drawn for initial pressures of 200 and 1200 lb. and maximum temperatures of 700 and 800 deg. fahr. In this cycle the steam is formed at constant pressure and is then considered to pass without any loss by throttling or radiation to the turbine, in which the adiabatic expansion, *cd*, takes place until exhaust pressure is reached as indicated by the state *d*. Condensation of the exhaust steam then takes place at constant

pressure as shown by the line *da*. This cycle has much to recommend it from the standpoint of simplicity and is the one in most common use today. It does not, however, have as great possibilities regarding thermal economy as some of the others.

9 *The Reheating Cycle* (B) is composed of the processes shown in Fig. 2. The steam expands adiabatically from *c* to *d* and then is reheated at constant pressure from *d* to *e*, after which a second adiabatic expansion takes place from *e* to *f*. The rest of the cycle is the same as cycle A. This reheating implies that the steam is

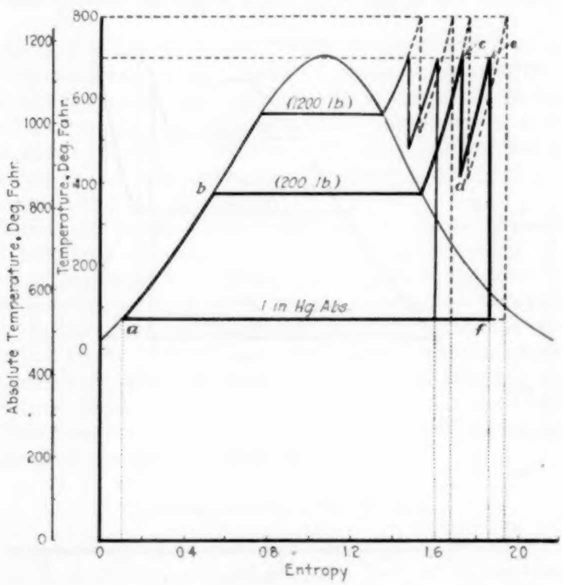

Fig. 2 Cycle B (Reheating)

For throttle pressure of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

carried back to the boiler plant to pass through a suitable reheater and is then returned to the turbine in which the expansion is completed, or else that there has been provided near the turbine a suitable separately fired reheating apparatus so that the steam does not have to be carried back to the boiler room. Both of these schemes involve certain complications and possibly difficulties in a real plant. The particular pressure at which the reheating should be carried out in order to give a maximum efficiency of this cycle is shown by curves which will be discussed later in connection with Figs. 8 and 9, and by the calculations given in Tables 9 and 10 of Appendix No. 1.[1] In these tables, cycle A is properly

[1] See note on page 711.

treated in the calculations as a special case of cycle B in which the reheating pressure is 100 per cent of the throttle pressure.

10 *The Cycle with Isothermal Superheating* (C) consists of the processes shown in Fig. 3. This name is used because the cycle involves an isothermal expansion of superheated steam as indicated by the line *cd*. It is not claimed that this process is likely to be approached even approximately in any real turbine, as it now seems that the expense and complication resulting from the attempt to supply heat to the steam while it is expanding in the turbine would probably more than offset the advantages to be

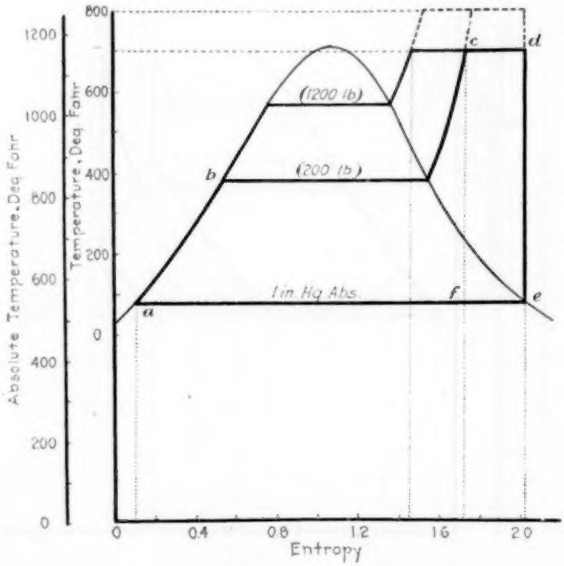

Fig. 3 Cycle C (Isothermal-Superheating)

For throttle pressures of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

gained thereby. The study of this cycle, however, does offer one very great advantage, in that it shows a maximum which might be approached by using an infinitely large number of constant-pressure reheating devices as explained for cycle B. The calculations for the efficiency of this cycle will be found in Table 11 of Appendix No. 1, and the efficiency curves are shown in Fig. 10.

11 *The Regenerative Cycle* (D) is made up of processes which may be indicated as in Fig. 4, although some of these processes are complex. This important cycle is called "regenerative" because steam is bled from the turbine at a number of stages to feedwater heaters, and the heat which is thus recovered is returned to the boiler.

12 In the ideal cycle an infinitely large number of bleeder heaters are assumed to be used, so that if superheated steam is not bled, the line *de* is drawn parallel to the liquid line *fa*. If there had been no bleeding, the adiabatic expansion would have continued to the state *e'*. On the other hand, with bleeding, successive portions of the steam are withdrawn from the turbine and condensed, heating the feedwater, while the remainder proceeds through the turbine, continuing the isentropic expansion.

13 If at each temperature the portion in the turbine and the portion which has been bled and condensed be considered to be

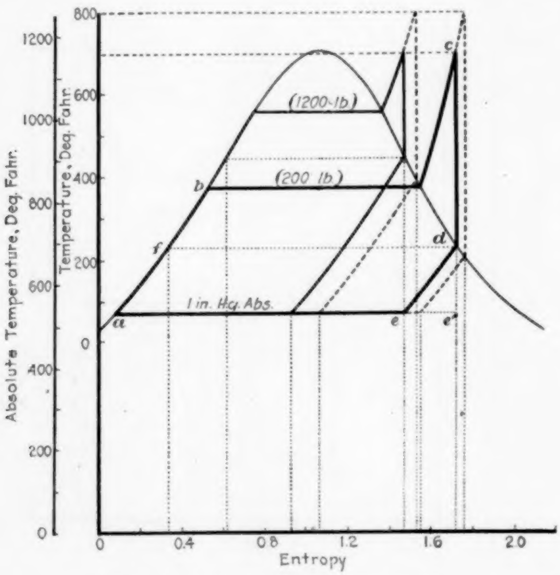

Fig. 4 Cycle D (Regenerative)

For throttle pressures of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

mixed together, the resultant state is that shown on a curve drawn parallel to the liquid line. In other words, the curve *de* is drawn for the entire weight of steam considered in the cycle and does not represent the state of only that portion of the steam continuing through the turbine. By drawing the diagram to represent all of the steam considered, the area *abcde* represents the energy available from this amount of steam and the equations for the cycle efficiency may be developed readily.

14 In case superheated steam should be bled the process cannot be shown in the superheated field by drawing a line parallel to the liquid line. The reason for this is that the specific heat is so much less for superheated steam than for water, that even though all of

the throttle steam should be bled there would not be enough heat available from it to heat the feedwater to the same temperature as that of the superheated steam at the beginning of bleeding. This statement does not mean that there are no cases in which it is economically desirable to bleed superheated steam, but it is intended to call attention to the fact that the temperature-entropy diagram, the equations, and the efficiency curves given in this paper are to be used only when saturated steam is bled.

15 It is particularly instructive to observe from Fig. 4 how the temperature of the feedwater rises in this cycle as we go to high

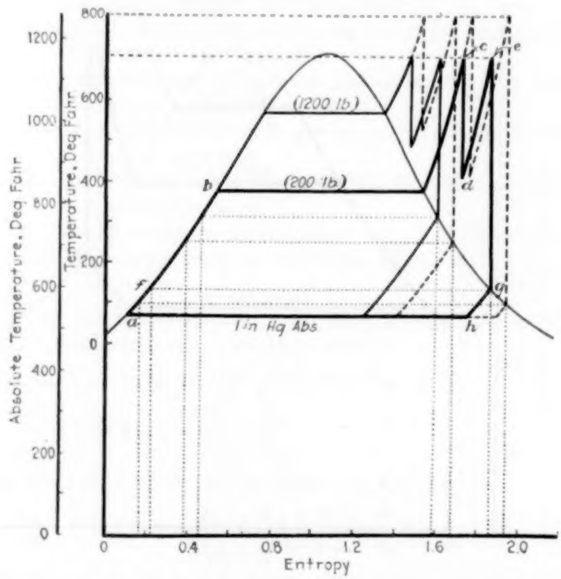

Fig. 5 Cycle E (Reheating-Regenerative)

For throttle pressures of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

pressures. For the ideal regenerative cycle using a throttle pressure of 1200 lb. per sq. in. and a temperature of 700 deg. the feedwater might be returned to the boiler at a temperature of about 450 deg., whereas for a throttle pressure of 200 lb. the feedwater temperature would be only about 225 deg.

16 The curves which show how the efficiency of this cycle is affected by varying the bleeding temperature may be seen in Figs. 11 and 12, and the calculations are given by Tables 12 and 13 in Appendix No. 1.

17 *The Reheating-Regenerative Cycle* (E) is outlined in Fig. 5. It is called "reheating-regenerative", for the reason that the steam is reheated as in the case of cycle B, and then the regenerative

principle is also applied as discussed for cycle D. Therefore the area *abcdegh* represents the available energy in B.t.u. per pound of throttle steam. This cycle implies considerable complication, but under certain conditions its possible advantages apparently justify the building of a plant to operate with this ideal. The results of the calculations for the efficiency of this cycle are given in Tables 14 and 15 in Appendix No. 1 and also in Figs. 13, 14, 15, and 16.

18 *The Isothermal-Regenerative Cycle* (F) is made up of the processes shown in Fig. 6. This cycle can hardly be called an

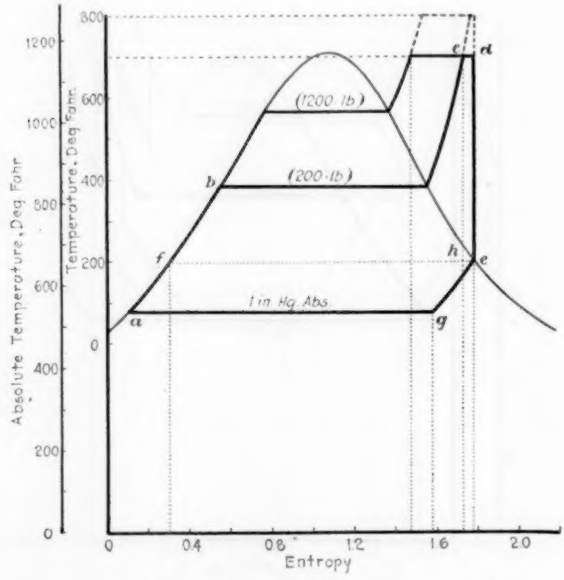

Fig. 6 Cycle F (Isothermal-Regenerative)

For throttle pressures of 1200 and 200 lb. per sq. in. abs., temperatures of 700 and 800 deg. fahr., and exhaust pressure of 1 in. Hg abs.

important one except in so far as it indicates the limiting possibilities of cycle E with an infinitely large number of reheaters, so that the isothermal expansion in the superheated region is approached as shown by the curve *cd*. The regenerative portion of this cycle is exactly similar to that described for the two previous cycles, except that the saturation temperature at which the bleeding begins is determined by the amount of isothermal expansion permitted in the turbine. In other words, the length of the line *cd* determines the temperature at *e* and consequently at *f*. Several different temperatures of bleeding have been used in the calculation of the efficiency of this ideal cycle and the results may be found in Tables 16 and 17 of Appendix No. 1, also in Fig. 17.

Efficiencies of Ideal Cycles

19 The efficiency of any ideal cycle is the ratio of the available energy, or the energy convertible into work, to the heat supplied during the entire cycle. In order to save space the cycle efficiencies will each be expressed in general terms, using the following notation:

Let H = specific total heat of the steam in B.t.u. per lb. for any state indicated by the subscript

q = specific total heat of the liquid in B.t.u. per lb. for any state indicated by the subscript

T = absolute temperature in deg. fahr. for any state indicated by the subscript

t = ordinary temperature in deg. fahr. for any state indicated by the subscript

p = absolute pressure in lb. per sq. in. for any state indicated by the subscript

V = specific volume in cu. ft. per lb. for any state indicated by the subscript

ϕ = specific entropy for any state indicated by the subscript

F = feed-pump work of the ideal cycle, in B.t.u. per lb. of water.

20 With this notation, and neglecting the feed-pump work — which is considered later — we then have:

For the *Rankine Cycle* (A) as shown by Fig. 1,

$$\text{Efficiency} = \frac{H_c - H_d}{H_c - q_a}$$

For the *Reheating Cycle* (B) as shown by Fig. 2,

$$\text{Efficiency} = \frac{H_c - H_d + H_e - H_f}{H_c - H_d + H_e - q_a}$$

For the *Cycle with Isothermal Superheating* (C) as shown by Fig. 3,

$$\text{Efficiency} = \frac{H_c - H_f + (t_d - t_e)(\phi_d - \phi_e)}{H_c - q_a + T_d(\phi_d - \phi_e)}$$

For the *Regenerative Cycle* (D) as shown by Fig. 4,

$$\text{Efficiency} = \frac{H_c - q_f - T_c(\phi_d - \phi_f)}{H_c - q_f}$$

For the *Reheating-Regenerative Cycle* (E) as shown by Fig. 5,

$$\text{Efficiency} = \frac{H_c - H_d + H_e - H_g + (t_g - t_h)(\phi_g - \phi_f)}{H_c - q_f + H_e - H_d}$$

For the *Regenerative Cycle with Isothermal Superheating* (F) as shown by Fig. 6,

$$\text{Efficiency} = \frac{H_c - H_h + (t_d - t_e)(\phi_d - \phi_e) + (t_f - t_g)(\phi_e - \phi_f)}{H_c - q_f + T_c(\phi_d - \phi_e)}$$

Work of the Boiler-Feed Pump in an Ideal Cycle

21 In calculating the efficiencies of ideal cycles, the work required to pump the water into the boiler is usually neglected. This concession to convenience will probably continue quite generally, since the cycle efficiency is only slightly reduced by including this term. However, since this item evidently increases in importance with the pressure, it has seemed worth while to compute it for the range of pressures considered in this paper, in order to give an accurate conception of its magnitude.

22 Calling the work of the ideal boiler-feed pump F, the true efficiency of each of the above cycles is found by subtracting from

TABLE 1 EFFECT OF FEED-PUMP WORK ON IDEAL CYCLE EFFICIENCY

Cycle	Throttle temperature, deg. fahr.	Boiler pressure, lb. per sq. in. abs.	Energy to operate ideal feed pump, = F in B.t.u. per lb.	Heat supplied neglecting F, = Q_1 in B.t.u. per lb.	Heat supplied including F, = $Q_1 - F$ in B.t.u. per lb.	Available energy neglecting F, = E in B.t.u. per lb.	Available energy including F, = $E - F$ in B.t.u. per lb.	Efficiency neglecting F, $e = E/Q_1$	Efficiency including F $e' = (E-F)/(Q_1-F)$	Difference in ideal-cycle efficiency, $e - e'$
A	700	200	0.6	1327.4	1326.8	447.7	447.1	0.3373	0.3370	0.0003
		400	1.2	1315.3	1314.1	481.4	480.2	0.3660	0.3654	0.0006
		600	1.8	1301.4	1299.6	497.3	495.5	0.3821	0.3813	0.0008
		800	2.4	1286.1	1283.7	505.0	502.6	0.3926	0.3915	0.0011
		1000	3.0	1269.9	1266.9	509.1	506.1	0.4009	0.3995	0.0014
		1200	3.6	1250.9	1247.3	508.0	504.4	0.4061	0.4044	0.0017
D Max.	700	200	0.6	1171.4	1170.8	429.0	428.4	0.3662	0.3659	0.0003
		400	1.2	1095.0	1093.8	446.5	445.3	0.4078	0.4071	0.0007
		600	1.8	031.6	1029.8	446.2	444.4	0.4325	0.4315	0.0010
		800	2.4	1974.2	971.8	439.2	436.8	0.4508	0.4495	0.0013
		1000	3.0	918.8	915.8	428.0	425.0	0.4658	0.4641	0.0017
		1200	3.6	863.3	859.7	411.9	408.3	0.4771	0.4749	0.0022

both the numerator and denominator of the equations given in Par. 20 the term F, whose value is given by the equation

$$F = \frac{144(p_b - p_a)V_a}{778} = \frac{144(p_b - 0.491)(0.0161)}{778}$$

$$= 0.00297(p_b - 0.491)\text{B.t.u. per lb.}$$

23 Table 1 shows the magnitude of this correction when applied to cycles A and D, and for the other cycles it would be only slightly different. Since this correction is relatively small, it affects the absolute value of the efficiency but little, and since its variations are still smaller, they affect only slightly the changes in efficiency due to changes in cycle or operating conditions; hence this correction will be neglected. However, in the consideration of the performance of real plants, the various losses in the feed piping, pump, and motor are such that the power required to operate the feed pumps becomes of considerable importance, especially at high pressures, as will be shown in Part II.

Steam Tables Used

24 In all of the calculations involving the properties of steam, Goodenough's tables were used, since his values are given for a pressure as high as 800 lb. per sq. in. in the superheated field. Beyond this pressure the values were calculated from Goodenough's equations. The numerical values of all quantities used in determining the various cycle efficiencies are given in Tables 9 to 17, inclusive, in Appendix No. 1. The efficiencies have been calcu-

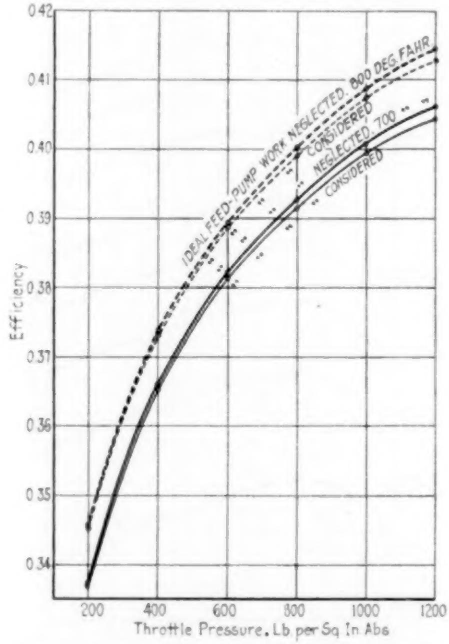

Fig. 7 Efficiency of Cycle A (Rankine)
(Throttle temperatures indicated on curves; exhaust pressure, 1 in. Hg abs.)

lated to a degree of accuracy not justified by our knowledge of the properties of steam when the absolute values of the efficiencies are considered. On the other hand, our study is concerned largely with relative values, or differences, and the only way to secure concordant results is to assume that our initial data are exact, and compute results to sufficient significant figures to bring out the differences. The differences between successive values are of course much more accurate than any of the absolute values.

Efficiency Curves

25 The effect of pressure on the efficiency of the Rankine cycle (A) is shown in Fig. 7 for the two throttle temperatures

of 700 and 800 deg. fahr., both with and without the correction for the ideal feed-pump work.

26 The curves in Fig. 7 show that increasing the pressure from 200 to 1200 lb. per sq. in. causes the efficiency of the Rankine cycle to increase about 20 per cent, while increasing the throttle temperature from a value of 700 to 800 deg. fahr. will cause the efficiency to be increased only about 2 per cent.

27 The efficiencies of the *Reheating Cycle* (B) for various

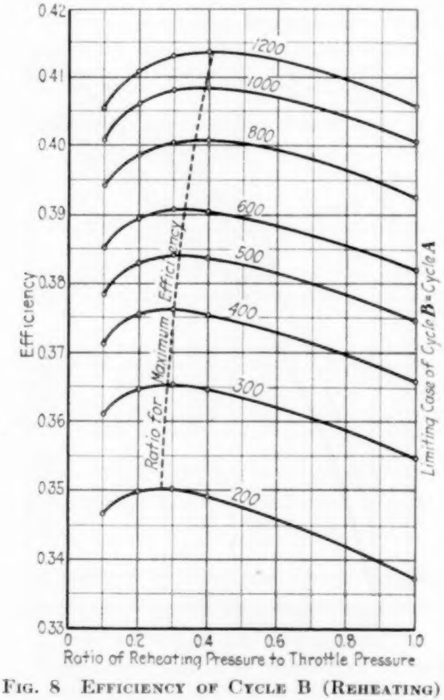

Fig. 8 Efficiency of Cycle B (Reheating)

Temperature at throttle, 700 deg. fahr.; exhaust pressure, 1 in. Hg abs.; throttle pressures (lb. per sq. in. abs.) indicated on curves.

throttle pressures and reheating pressures are shown by the curves in Fig. 8. The increase in the throttle pressure from 200 to 1200 lb. per sq. in. will cause the efficiency of the cycle to increase by about the same percentage as the Rankine, which may be considered as merely a limiting case of cycle B, with the reheating pressure equal to the throttle pressure.

28 The points of maximum efficiency for each throttle pressure have been connected by a dotted line so that for any throttle pressure the proper reheating pressure to give the maximum cycle efficiency is readily determined. This maximum-efficiency curve

is based on ideal conditions and is to be used only as one important factor in trying to select the *best* reheating pressure for a real plant; throttling, the cost of piping, and the quality of the steam going to the lower stages of the turbine must also be considered.

29 For a throttle temperature of 800 deg. fahr. and the reheating cycle the curves shown in Fig. 9 should prove helpful in the same manner as those just discussed for 700 deg.

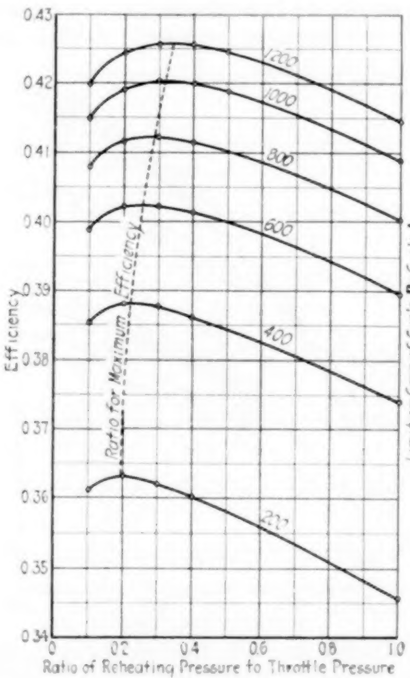

FIG. 9 EFFICIENCY OF CYCLE B (REHEATING)

Temperature at throttle, 800 deg. fahr.; exhaust pressure, 1 in. Hg abs.; throttle pressures (lb. per sq. in. abs.) indicated on curves

30 For the *Isothermal-Superheating Cycle* (C) the curves of efficiency for the two throttle temperatures are shown in Fig. 10. The curves of cycle C for both temperatures are appreciably higher than those for cycle A or B for the same temperatures, a result which is to be expected, since in cycle C a greater portion of the heat supplied during the cycle is added at the maximum temperature. If there were available no cycles other than A, B, and C, whose relative efficiencies are clearly so favorable to C, one might attempt the difficult task of making a turbine in which the isothermal superheating process is approached; but fortunately

there is also the regenerative cycle (D), which has a higher efficiency than C for sufficiently high throttle pressures. Fig. 10 shows that cycle D is superior above 450 lb. with an initial temperature of 700 deg. fahr. and above 750 lb. with an initial temperature of 800 deg. fahr. Further, cycle D merely calls for bleeder heaters, which are simple and not very costly, while C requires a new type of turbine built so as to maintain isothermal expansion.

31 Considering the *Regenerative Cycle* (D) alone, it should be observed from Fig. 11 how the efficiency varies with a change in bleeding temperature (or pressure), throttle pressure, and

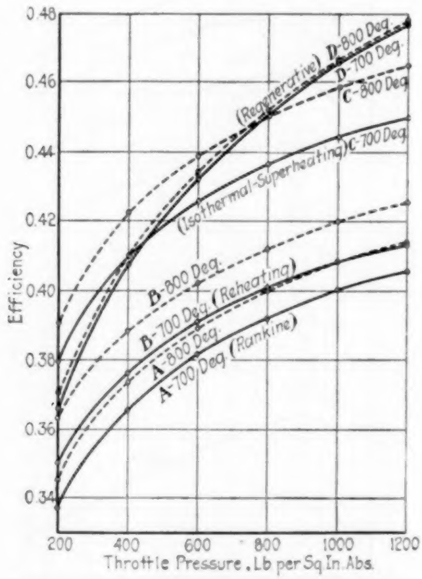

Fig. 10 Comparative Maximum Efficiencies of Cycles A, B, C, and D

Throttle temperatures of 700 and 800 deg. fahr. and exhaust pressure of 1 in. Hg abs.

temperature. For each case it may be seen that the maximum efficiency occurs when the bleeding begins as soon as the saturation line is reached. The reason for the very slight improvement in efficiency resulting from increasing the throttle temperature to 800 deg. fahr., is that as the throttle temperature is increased the maximum bleeding temperature goes down, as shown by the diagram of Fig. 4. The rising initial temperature tends to increase the cycle efficiency; but the decreasing boiler-feed temperature materially offsets this effect. The relation between throttle pressure and temperature, and the bleeding temperature to give maximum efficiency of this cycle, is shown by the curves in Fig. 12. The question of how close we may be able to approach these values

in a real plant will be considered in the second part of this paper. Attention is called to the fact that no study of this cycle with bleeder heating at temperatures above saturation is included. This phase of the subject will be referred to later.

32 The curves which represent the efficiency of the *Reheating-Regenerative Cycle* (E) are shown in Figs. 13, 14, 15, and 16. Fig. 13 shows how the efficiency is affected by changing the ratio of reheating pressure to throttle pressure, and it also shows that there is a much greater improvement produced by increasing the throttle

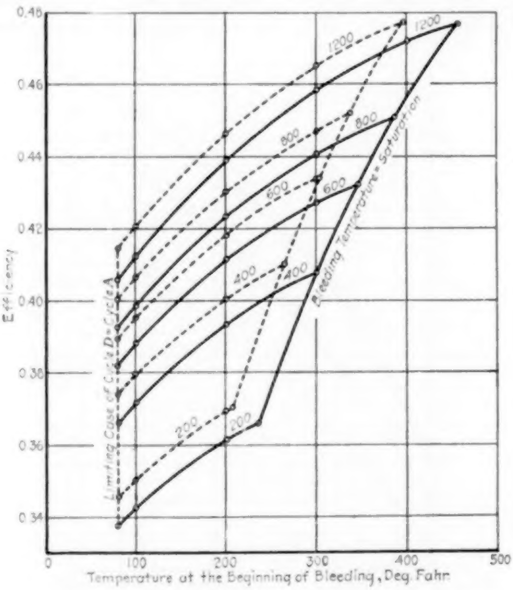

FIG. 11 EFFICIENCY OF CYCLE D (REGENERATIVE)

Throttle pressures (lb. per sq. in. abs.) indicated on curves. Throttle temperature of 700 deg. fahr. shown by solid line; of 800 deg. fahr. by broken line. Exhaust pressure, 1 in. Hg abs.

temperature to 800 deg. fahr., when this ratio is small. The temperatures of saturated steam at the beginning of bleeding, which change with the various throttle pressures and reheating pressures, are shown by the curves in Figs. 14 and 15 for the throttle temperatures of 700 and 800 deg. fahr., respectively. Fig. 16 shows the curves of Figs. 14 and 15 combined, except for the omission of the reheating pressure ratio. In all of the curves representing the efficiency of cycle E, it may be noted that only at the lower throttle pressure does this cycle give a higher efficiency than the regenerative cycle (D).

33 Cycle F (*Isothermal-Regenerative*) gives efficiencies with values which vary from those of cycle C to those which are equal

to the maximum of cycle D, as may be seen from Fig. 17. In other words, the ratio of isothermal expansion in the ideal turbine, V_d/V_e of Fig. 6, has been varied from unity which corresponds to cycle D, to the maximum which corresponds to cycle C. This maximum ratio of expansion is arbitrarily taken such that at the end of the ideal adiabatic expansion the steam is exactly dry and saturated at the exhaust pressure, so that for this case there can be no bleeding.

34 In Fig. 18 are given the efficiencies of the six cycles arranged so that they may be compared for a throttle temperature of 700 deg. fahr. and various throttle pressures from 200 to 1200 lb. per sq. in. Clearly, cycle D is the one which deserves first place,

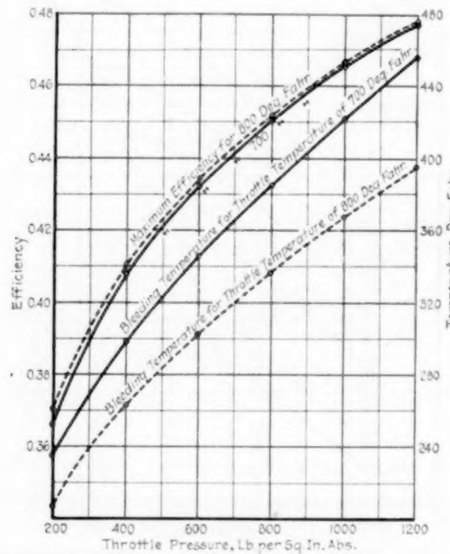

Fig. 12 Maximum Efficiency and the Corresponding Bleeding Temperature of Cycle D (Regenerative)

especially for the higher pressures, as far as efficiencies on the ideal basis are concerned.

35 In addition to the efficiency, one other value which is important to engineers and which may be calculated for the ideal cycles is the amount of energy available for work in the ideal turbine per unit volume of the steam at exhaust. This ratio is important because it is a partial measure of the relative sizes and costs of the turbines required to give the same power when operating on the different cycles. The superiority of the regenerative cycle in this regard also, is shown by Fig. 19. Cycles C and F were not considered of sufficient importance to justify drawing their curves in this figure. Their values were calculated, however, and may be found in Tables 11, 16, and 17 of Appendix No. 1.

Conclusions From Part I

36 For any cycle herein studied it is clear that an increase in pressure causes an increase in efficiency and in energy available per unit volume of exhaust steam. This increase in efficiency with the pressure is less marked in the simple Rankine cycle than in the ones with more complicated apparatus.

37 Changing the throttle temperature from 700 to 800 deg. fahr. causes a very marked increase in the efficiency of all cycles herein studied except those using the regenerative principle, which are improved but very little by this increased temperature.

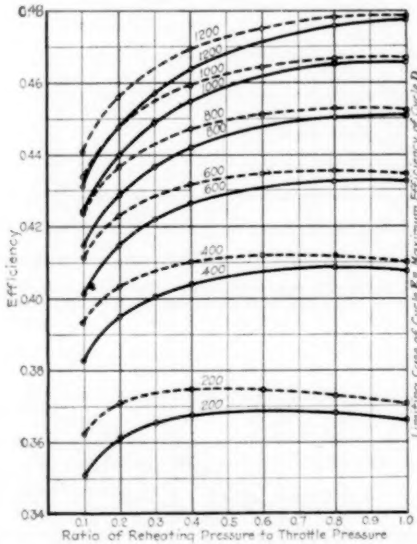

Fig. 13 Efficiency of Cycle E (Reheating-Regenerative)

Throttle pressures (lb. per sq. in. abs.) indicated on curves. Throttle temperatures of 700 and 800 deg. fahr. respectively shown by solid and broken lines. Exhaust pressure, 1 in. Hg abs.

38 From the consideration of the ideal cycles alone it appears that the Rankine cycle, which is the one requiring the least apparatus, is the least efficient throughout the entire range of pressures, and yields the least energy per unit volume of exhaust steam. At the top of the list, throughout the upper portion of the pressure range considered, and very close to the top in the lower portion, stands the regenerative cycle both as to efficiency and available energy per unit volume of exhaust steam. The extra apparatus which must be used in order to follow the regenerative cycle instead of the Rankine would seem to be simple enough to justify the extra expense in many cases. The study of the modifying factors in the real plant are treated in Part II.

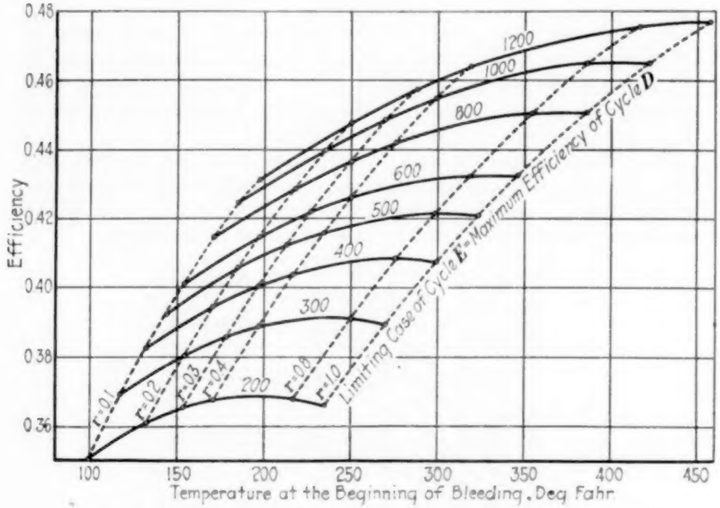

Fig. 14 Efficiency of Cycle E (Reheating-Regenerative)

Temperature at throttle, 700 deg. fahr. Exhaust pressure, 1 in. Hg abs. Throttle pressures (lb. per sq. in. abs.) indicated on curves. r = ratio of reheating pressure to throttle pressure.

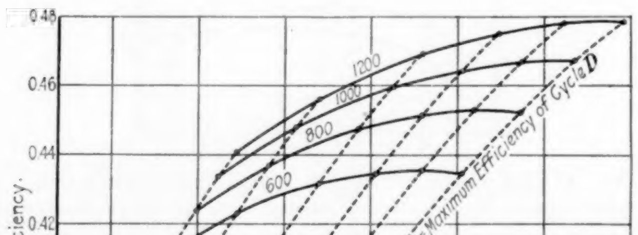

Fig. 15 Efficiency of Cycle E (Reheating-Regenerative)

Temperature at throttle, 800 deg. fahr. Exhaust pressure, 1 in. Hg abs. Throttle pressures (lb. per sq. in. abs.) indicated on curves. r = ratio of reheating pressure to throttle pressure.

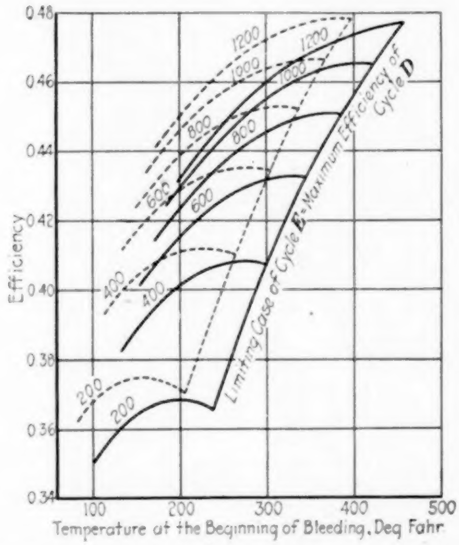

FIG. 16 EFFICIENCY OF CYCLE E (REHEATING-REGENERATIVE)

Exhaust pressure, 1 in. Hg abs. Throttle pressures (lb. per sq. in. abs.) indicated on curves. Throttle temperatures of 700 and 800 deg. fahr. respectively shown by solid and broken lines.

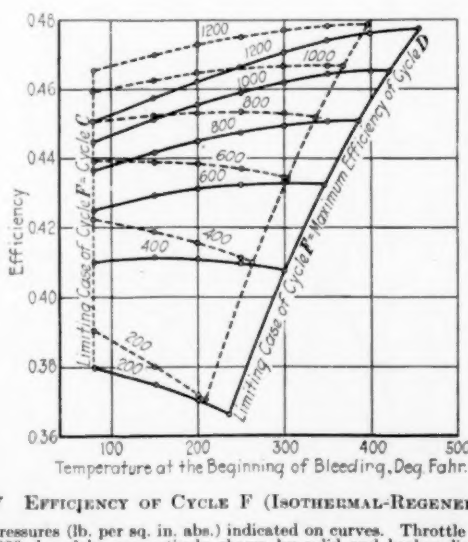

FIG. 17 EFFICIENCY OF CYCLE F (ISOTHERMAL-REGENERATIVE)

Throttle pressures (lb. per sq. in. abs.) indicated on curves. Throttle temperatures of 700 and 800 deg. fahr. respectively shown by solid and broken lines. Exhaust pressure, 1 in. Hg abs.

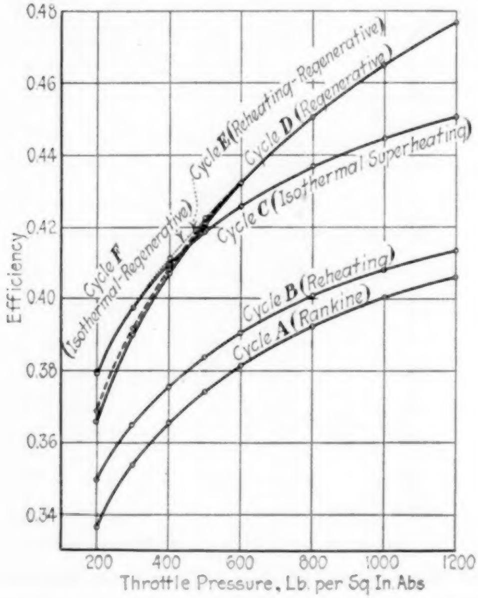

FIG. 18 COMPARATIVE MAXIMUM EFFICIENCIES OF CYCLES A, B, C, D, E, AND F
(Throttle temperature of 700 deg. fahr. and exhaust pressure of 1 in Hg abs.

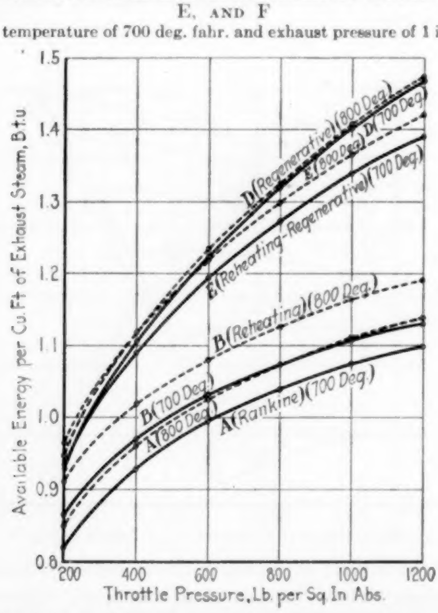

FIG. 19 AVAILABLE ENERGY PER CUBIC FOOT OF EXHAUST STEAM
(Throttle temperatures indicated on curves. Exhaust pressure, 1 in Hg abs.)

II — PROBABLE RESULTS FROM REAL PLANTS

Method of Estimating Plant Performance

39 The efficiency of each of the ideal cycles may be calculated so that the results are correct except as they may be slightly affected by uncertainty in the values of the properties of high-pressure steam. The real steam plant, however, can never do as well as its corresponding ideal, and the degree of approach made toward this ideal is a very proper measure of excellence of the design, construction, and operation of such a plant. In a study of this kind it is desirable to determine as carefully as possible the probable performance of real plants, even though such plants may never be built to operate on some of the cycles discussed in this paper.

40 Performance figures for a plant which has not been built may be obtained by modifying the results calculated for the ideal cycle on which the plant is intended to operate. The ideal-cycle output per unit mass of steam, multiplied by the engine efficiency and by the ratio of the heat supplied per pound of steam in the actual case to that in the corresponding ideal gives the gross generator output. With the gross generator output per unit mass of steam the flow of steam necessary to produce any gross power output is ascertained. From this gross output must be subtracted the power required by all of the plant auxiliaries which, for the convenience of this study, were assumed to be electrically driven. This power may be estimated rather closely for the plant load and boiler rating which are considered proper for a study of this kind. Any steam required for steam-driven auxiliaries, losses, etc., could be estimated per pound of steam to the main units and thus the total steam per unit of net plant output determined, in case it should be desired to use steam-driven auxiliaries. The heat required to generate the necessary steam depends upon the boiler-room efficiency, which may be assumed to be the same for all cases in a comparative study of the type here under consideration, the extent of the heating surface being adjusted to give the efficiency assumed. The efficiency corresponding to the most economical heating surface might also be worked out, but the labor is very arduous when so many cycles and pressures are involved.

41 The cost of the complete plant can be estimated only by finding out as carefully as time permits the cost of the various pieces of apparatus which make up the entire plant for each case considered.

Probable Engine Efficiencies

42 Of all the power-plant apparatus involved in a cycle the turbine group is the one whose engine efficiency is most sensitive to changes in the cycle and in the pressure and temperature of the steam. Instead of dealing with the efficiencies of turbine, generator

and heaters separately it is more convenient to work with the combined group. The engine efficiency of this group is the ratio of its thermal efficiency to the thermal efficiency of the ideal cycle.

43 The engine efficiencies to be expected from a turbo-generator of 30,000 kw. capacity and its heaters when operating on the cycles under consideration have been calculated as well as possible, and the results are shown in Fig. 20.

44 The method of establishing these curves will now be discussed, and in this connection it should be noted that the use of different steam tables will give slightly different values.

45 Many tests have been made on large turbo-generator units operating with steam at pressures of from 200 to 300 lb. per sq. in. so that the A or Rankine curve is well established within this

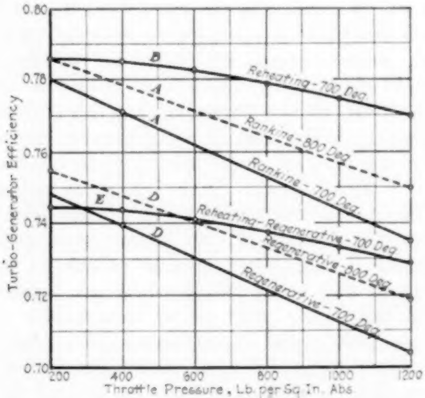

FIG. 20 PROBABLE ENGINE EFFICIENCIES OF THE TURBO-GENERATOR AND HEATERS

(For 30,000-kw. units and exhaust pressure of 1 in. Hg abs.)

low-pressure range. This curve, for use with the Rankine cycle, was extended to the higher pressures by allowing for the ill effect of the extra moisture encountered as the pressure increases. It is drawn as a straight line with a slope that represents substantially the results as obtained by Baumann and others who have made this calculation.

46 For curve B, the values were established in the following manner: Neglecting the throttling effect through the reheater and its piping, the extra superheat from the reheater was estimated to increase the efficiency of the A-curve from a value of 0.771 to 0.796 at 400 lb. and from 0.735 to 0.775 at 1200 lb. pressure. A straight line was then drawn through these two points, and from this line was subtracted the calculated reduction in efficiency due to throttling through the reheater and its piping. This loss of pressure

due to throttling was taken as 3 per cent of the reheating pressure for the piping and 5 lb. per sq. in. for the reheater. Table 2 shows the method of estimating the change in efficiency caused by the loss of pressure during the entire reheating process.

47 The values given in the last column are those used to give curve B of Fig. 20. This curve is intended to present consistent values for the purpose of this study, and it is not intended to be used for power-plant design without further study. The reheater may, very naturally, be made to have either more or less pressure drop through it, and evidently the drop through the piping may be made almost anything desired from a consideration of cost. Furthermore, the reheating pressure may often be other than the 30 per cent of throttle pressure which was used in this table and in drawing curve B. As the reheating pressure is increased to values very much higher than 30 per cent of the

TABLE 2 EFFECT OF THROTTLING IN CYCLE B

Pressures, lb. per sq. in. abs.		Pressure Drop, lb. per sq. in.			Available Energy, B.t.u. per lb.			Probable Engine Efficiency	
Throttle	Reheating	Through reheating piping, 3 per cent	Through reheater	Total	Lost by throttling	Total for ideal cycle	Per cent lost by throttling	For no throttling	Including throttling
1200	360	10.8	5	15.8	3.6	592.0	0.61	0.775	0.769
800	240	7.2	5	12.2	4.3	579.7	0.74	0.786	0.779
600	180	5.4	5	10.4	4.7	569.1	0.83	0.791	0.783
400	120	3.6	5	8.6	5.8	549.9	1.06	0.796	0.785
200	60	1.8	5	6.8	8.4	513.6	1.64	0.802	0.786

throttle pressure, the ideal-cycle efficiency changes as shown by Fig. 8, and curve B of Fig. 20 approaches curve A. Or in other words, the reheating cycle becomes the Rankine when the reheating pressure becomes 100 per cent of the throttle pressure.

48 On the other hand, if the reheating pressure is decreased a limited amount there will naturally be an improvement in the efficiency of the low-pressure sections of the turbine. Whether the economy of the turbine as a whole will be improved by this reduction of reheating pressure depends upon how much the heat to the exhaust has been increased relatively to the net change in stage efficiency of entire turbine. It is apparent, then, that the reheating pressure should be determined rather carefully when actual designs are under consideration. Attention is called to these details for the purpose of indicating that plants using different pressures may be expected to show considerable variation from curve B. It is also worth while to note how very serious is the

throttling loss in reheater equipment at the lower initial pressures as compared to the higher ones.

49 For cycle C there is no efficiency curve drawn, as we shall very probably never see a plant built to operate on this cycle. The difficulties of approaching the isothermal expansion in the turbine appear at this time to make the first cost of such a machine very much more than is justified by the best possible expectations regarding its thermal economy.

50 For curve D, or the one to be used with the regenerative

TABLE 3 ESTIMATED TEMPERATURE OF FEEDWATER ENTERING ECONOMIZER IN REGENERATIVE PLANT WITH THROTTLE TEMPERATURE OF 700 DEG. FAHR.

Throttle pressure, lb. per sq. in. abs.	200	400	600	800	1000	1200
Superheat at throttle, deg. fahr.	318	255	213	181	155	132
Entropy at throttle in ideal cycle	1.7239	1.6387	1.5839	1.5411	1.5039	1.4707
Maximum temperature at beginning of bleeding in ideal cycle, deg. fahr.	235	299	346	386	422	457
Energy available before bleeding begins in ideal cycle (E_1), B.t.u. per lb.	214.2	181.6	156.2	134.2	114.8	95.8
Total available energy in ideal cycle (E_t), B.t.u. per lb.	428.9	446.7	446.3	439.2	427.5	412.0
Ratio, E_1/E_t	0.50	0.41	0.35	0.31	0.27	0.23
Probable increase in entropy to saturation point, actual turbine, per cent	3.1	2.8	2.5	2.3	2.05	1.8
Probable entropy at saturation point, actual turbine	1.7773	1.6846	1.6235	1.5765	1.5347	1.4972
Corresponding saturation temperature, deg. fahr.	201	236	311	353	392	430
Corresponding saturation pressure, lb. per sq. in. abs.	11.8	37.3	78.8	140	225	343
Estimated pressure drop through valve and pipe to high-pressure heater, lb. per sq. in.	1.8	3.9	5.9	7.0	8.0	8.8
Estimated loss of temperature in actual plant compared to ideal, due to:						
(*a*) Reheating in turbine	34	36	35	33	30	27
(*b*) Pressure drop through valve and pipe to high-pressure heater	8	6	4	4	2	2
(*c*) Temperature difference between steam and water in high-pressure heater	5	5	5	5	5	5
(*d*) Radiation loss in feed line	(Less than 0.5 deg. fahr.)					
Total estimated loss of temperature, deg. fahr.	47	47	44	42	37	34
Probable temperature of water to economizer, deg. fahr.	188	252	302	344	385	423

cycle, a value of 74 per cent is believed to be a fair one for a pressure of 400 lb., as it was obtained from careful calculations based upon actual guarantees. Through this point a straight line was drawn parallel to the one for the Rankine. This latter step is possibly somewhat unfair to plants designed to operate on this cycle at the higher pressures, as the regenerative feed-heating system may be expected to give feed temperatures which approach closer to their corresponding ideal maximum feed temperatures as we go to higher throttle pressures. This last statement is based upon two facts: First, for any given throttle temperature the higher the throttle pressure is, the less will be the reheating in any

given turbine before the saturation state is reached; secondly, the loss of available temperature head by reason of the throttling through the high-pressure bleeder valve and piping, will be considerably reduced because of the higher bleeder pressures used with the high throttle pressures. The magnitude of these factors is indicated by Table 3, which was calculated for the purpose of determining the probable feed temperatures. It should be noted that the use of a larger number of bleeder heaters is implied as the

TABLE 4 ESTIMATED TEMPERATURE OF FEEDWATER ENTERING ECONOMIZER IN REHEATING-REGENERATIVE PLANT WITH THROTTLE TEMPERATURE OF 700 DEG. FAHR.

Throttle pressure, lb. per sq. in. abs.	200	400	600	800	1000	1200
Reheating pressure, lb. per sq. in. abs.	80	160	240	320	400	480
Maximum temperature at beginning of bleeding, ideal cycle, deg. fahr.	170.9	217.6	250.0	276.3	298.8	319.0
Entropy at the end of reheating, ideal cycle	1.8294	1.7501	1.7022	1.6670	1.6387	1.6147
Drop of pressure through heater, lb. per sq. in.	5.0	5.0	5.0	5.0	5.0	5.0
Drop of pressure through pipes (3 per cent), lb. per sq. in.	2.4	4.8	7.2	9.6	12.0	14.4
Total reheating pressure drop, lb.	7.4	9.8	12.2	14.6	17.0	19.4
Entropy after throttling, reheater	1.8409	1.7573	1.7079	1.6721	1.6437	1.6193
Probable increase in entropy to saturation point, actual turbine, per cent	3.4	3.2	3.0	2.9	2.8	2.7
Probable entropy at saturation point, actual turbine	1.9034	1.8134	1.7591	1.7206	1.6897	1.6629
Corresponding saturation temperature, deg. fahr.	134	180	212	237	259	279
Corresponding saturation pressure, lb. per sq. in. abs.	2.47	7.5	14.7	23.6	34.8	48.5
Estimated pressure drop through valve and pipe to high-pressure heater, lb. per sq. in.	0.5	1.2	2.0	2.9	3.8	4.6
Estimated loss of temperature in actual plant compared to ideal, due to:						
(a) Throttling in reheating system and reheating in turbine	37	38	38	39	40	40
(b) Pressure drop through valve and pipe to high-pressure bleeder heater	8	8	7	7	6	6
(c) Temperature difference between steam and water in high-pressure heater	5	5	5	5	5	5
(d) Radiation loss in feed line		(Less than 0.5 deg. fahr.)				
Total estimated loss of temperature, deg. fahr.	50	51	50	51	51	51
Probable temperature of feedwater to economizer, deg. fahr.	121	167	200	225	248	268

throttle pressure is increased. The number of bleeder heaters used for this study is given in Table 6.

51 For curve E, or the one to be used with the reheating-regenerative cycle having a reheat pressure of 40 per cent of the throttle pressure, it seemed proper to base values upon both B and D, since this curve is to represent the degree of approach toward an ideal involving both reheating and regenerating. As the bleeding pressure for cycle E is necessarily lower than that for cycle D, each using the same throttle pressure, it naturally follows from what was discussed in the previous paragraph that the feed

temperature actually attained by a real plant following this cycle will not approach its corresponding ideal temperature as closely as in D, as is shown in Table 4. This difference was estimated to be sufficient to make a reduction of about 1 per cent in the engine efficiency. Curve E was therefore drawn at the same distance below curve B that D less 1 per cent is below A. Curve E is probably less reliable than any of the others, as it involves more complex relations and had to be drawn without any check point available from test. For reheating pressures much higher than 40 per cent of the throttle pressure, curve E will approach D as its limiting case, which is for no reheating at all. The ideal cycle efficiencies vary with the reheat pressure as shown in Fig. 13.

52 No curve has been drawn to represent probable engine efficiencies of the group operating on cycle F, for the same reason as given for the omission of those relating to cycle C.

53 Only cycles A and D were considered when estimating what may be expected with an initial temperature of 800 deg. fahr. It was assumed that a 100 per cent increase in the degree of superheat at the throttle would cause an increase of 2 per cent in the turbine efficiency. Increasing the throttle temperature from 700 deg. to 800 deg. therefore means increasing the superheat from 318 deg. to 418 deg. or 31.5 per cent for a pressure of 200 lb., and from 132 deg. to 232 deg. or 75.7 per cent for a pressure of 1200 lb. per sq. in. The corresponding increase in turbine efficiency would therefore be 0.6 per cent at 200 lb. and 1.5 per cent at 1200 lb. With these corrections applied for the two extremes of pressure, straight lines were then drawn as shown by the broken lines for cycles A and D at 800 deg. For cycles B and E the improvement in plant economy which may be expected by increasing the throttle temperature to 800 deg. would surely be less than for cycles A and D, but it does not seem worth while at this time to try to estimate values for these additional curves.

Condenser Surface

54 The flow of steam necessary to produce a gross output of 30,000 kw. for each pressure and the four cycles, is shown in Table 5. Using the notation there given, the flow of steam in pounds per hour, $W_s = 3413 \times 30{,}000/E_a$. The efficiency of the generator was assumed to be 97 per cent, and 1 per cent of the available energy was considered a fair value for the energy lost by radiation and to operate the turbine oil pumps. With these assumptions the heat to the condenser was calculated and the results are given in Table 5.

55 The amount of cooling surface required for the condenser was based on a heat-transfer rate of 470 B.t.u. per sq. ft. of surface per hr. per deg. fahr. mean temperature difference. A fair mean temperature difference between the water and the steam was believed to be 15 deg. fahr. (See Table 5 and Fig. 21.)

Auxiliary Power

56 The power required by the auxiliaries was estimated to be as given in Table 6. From this table may be found the values which were considered reasonable for the losses in the steam and feed lines. With these losses, and with the pump and motor efficiencies as given, the power required for the boiler-feed and hotwell pumps was found. The feed-pump efficiency was taken as 70 per cent for all pressures. This value is probably too high rather than too low, but so long as it was used for each cycle the comparative results are not seriously affected.

57 The power needed by the motors to operate the circulating

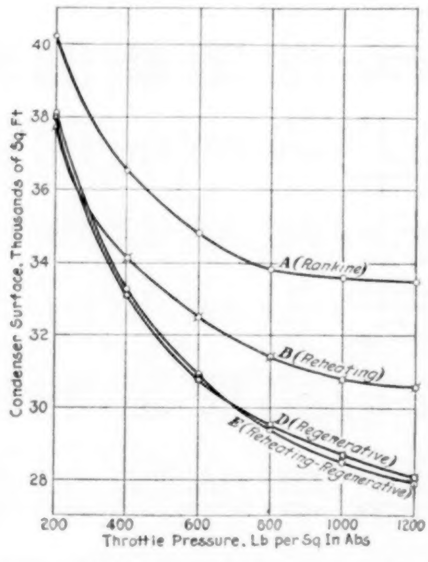

Fig. 21 Estimated Condenser Surface for a 30,000-Kw. Turbine
(Throttle temperature, 700 deg. fahr. Exhaust pressure, 1 in. Hg abs.)

pumps was determined by assuming a total friction head of 12 ft., a pump efficiency of 80 per cent, and a motor efficiency of 85 per cent. The power for the stokers and forced draft was taken proportional to the fuel burned. The power for the induced-draft fans was based upon the draft loss and the flow of gas which corresponds to a boiler rating of 160 per cent.

58 The calculations were made for the four cycles and the four pressures as indicated by the curves of Fig. 22, which show the power required by each auxiliary motor. These curves also show very clearly how the power required by the boiler-feed pumps rises

TABLE 5 CALCULATIONS FOR ESTIMATING STEAM FLOW, CONDENSER SURFACE, AND CIRCULATING WATER

Cycle	Throttle pressure, lb. per sq. in. abs.	Heat supplied in B.t.u. per lb. of steam, ideal cycle (from Table 5 A) Q_i	Ideal cycle efficiency (from Part I) e_i	Engine efficiency of turbo-generator and heaters (from Figure 20) e_r	Thermal efficiency of actual turbo-generator and heaters $e_a = e_i e_r$	Estimated feed temperature to economizer, actual plant (from Tables 3 & 4 for D & E) t_f	Heat supplied in B.t.u. per lb. of steam, actual turbine (from Table 5A) Q_a	Energy delivered by generator in B.t.u. per lb. of steam to throttle $E_a = e_a Q_a$	Gross output of main generator, kw. K_g	Steam flow to turbine, lb. per hr. $W_s = \frac{3413K_g}{E_a}$	Estimated heat to condenser, millions of B.t.u. per hr. $Q_c = W_s\left[\left(1-\frac{e_a}{.97}\right)Q_a - .01A.E.\right]$	Condenser surface, sq. ft. $\frac{\text{Heat to condenser}}{15 \times 470}$	Circulating water, thousands of lb. per hr. $\frac{\text{Heat to condenser}}{10 \times 1000}$
A	200	1327	0.337	0.780	0.263	75	1331	350	30,000	293,000	283	40,200	28,300
B (0.3)	200	1467	0.350	0.786	0.275	75	1449	398	30,000	257,500	266	37,700	26,600
D (Max.)	200	1171	0.366	0.748	0.274	188	1218	333	30,000	308,000	268	38,000	26,800
E (0.4)	200	1346	0.368	0.745	0.273	121	1378	376	30,000	273,000	269	38,100	26,900
A	400	1315	0.366	0.771	0.282	75	1319	372	30,000	276,000	257	36,500	25,700
B (0.3)	400	1461	0.376	0.785	0.295	75	1443	426	30,000	240,000	240	34,100	24,000
D (Max.)	400	1095	0.408	0.740	0.302	252	1142	344	30,000	298,000	233	33,100	23,300
E (0.4)	400	1293	0.404	0.744	0.301	167	1326	399	30,000	257,000	234	33,200	23,400
A	600	1301	0.382	0.762	0.291	75	1305	380	30,000	270,000	245	34,800	24,500
B (0.3)	600	1455	0.391	0.783	0.306	75	1436	439	30,000	234,000	229	32,500	22,900
D (Max.)	600	1032	0.433	0.731	0.316	302	1077	340	30,000	301,000	217	30,800	21,700
E (0.4)	600	1253	0.426	0.741	0.316	200	1286	406	30,000	253,000	218	30,900	21,800
A	800	1286	0.393	0.753	0.296	75	1290	382	30,000	268,000	238	33,800	23,800
B (0.3)	800	1448	0.401	0.779	0.312	75	1431	446	30,000	229,000	221	31,400	22,100
D (Max.)	800	974	0.451	0.722	0.325	344	1018	331	30,000	310,000	208	29,500	20,800
E (0.4)	800	1219	0.442	0.738	0.327	225	1255	411	30,000	250,000	207	29,400	20,700
A	1000	1270	0.401	0.744	0.298	75	1274	380	30,000	270,000	237	33,600	23,700
B (0.3)	1000	1442	0.406	0.775	0.316	75	1425	451	30,000	227,000	217	30,800	21,700
D (Max.)	1000	919	0.466	0.713	0.332	385	959	318	30,000	322,000	202	28,700	20,200
E (0.4)	1000	1189	0.455	0.734	0.334	248	1224	409	30,000	251,000	201	28,500	20,100
A	1200	1251	0.406	0.735	0.299	75	1255	375	30,000	273,000	236	33,500	23,600
B (0.3)	1200	1433	0.413	0.770	0.318	75	1418	451	30,000	228,000	216	30,600	21,600
D (Max.)	1200	863	0.477	0.704	0.336	423	900	302	30,000	339,000	198	28,100	19,800
E (0.4)	1200	1159	0.464	0.729	0.338	268	1195	404	30,000	254,000	197	27,900	19,700

* Throttle temperature, 700 deg. fahr. Exhaust pressure, 1 in. Hg abs.

TABLE 6 ESTIMATED AUXILIARY POWER REQUIRED FOR 30,000 KW. GROSS OUTPUT FROM MAIN UNIT

	Cycle A (Rankine)				Cycle B (Reheating)				Cycle D (Regenerative)				Cycle E (Reheating-Regenerative)			
Throttle pressure, lb. per sq. in. abs.	200	400	800	1200	200	400	800	1200	200	400	800	1200	200	400	800	1200
Pressure drop in main, lb. per sq. in.	5	12	36	58	3	7	14	21	6	15	31	51	4	9	18	27
Pressure drop in superheater, lb. per sq. in.	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Boiler pressure, lb. per sq. in. abs.	210	417	841	1263	208	412	819	1226	211	420	836	1256	209	414	823	1232
Number of bleeder heaters	0	0	0	0	0	0	0	0	2	3	3	4	1	2	2	3
Friction loss in heaters, lb. per sq. in.	0	0	0	0	0	0	0	0	20	30	30	40	10	20	20	30
Friction loss in feed pipe and lift, lb. per sq. in.	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45
Friction loss in economizer, lb. per sq. in.	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Friction loss in regulating valve, lb. per sq. in.	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Total pressure to be overcome by hotwell and boiler-feed pumps, lb. per sq. in.	285	492	916	1338	283	487	894	1301	306	525	941	1371	294	509	918	1337
Steam flow, thousand lb. per hr	293.0	276.0	268.0	273.0	257.5	240.0	229.0	228.0	308.0	298.0	310.0	339.0	273.0	257.0	250.0	254.0
Hotwell:																
Pump delivery pressure, lb. per sq. in. abs.	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Pump and motor efficiency (0.51 × 0.85)	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43	0.43
Power, kilowatts	30	28	27	28	26	24	23	23	31	30	32	35	28	26	25	25
Boiler-feed pumps:																
Delivery pressure, lb. per sq. in.	235	442	866	1288	233	437	844	1251	256	475	891	1321	244	459	868	1287
Pump and motor efficiency (0.70 × 0.85)	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60
Power, kilowatts	109	193	368	556	95	166	305	454	125	224	436	708	105	186	343	515
Power supplied to motors:																
For stokers, kilowatts	12	11	10	10	11	10	10	10	12	11	10	10	12	11	11	11
For forced draft, kilowatts	87	81	78	77	83	77	74	72	94	86	79	73	84	76	69	68
For induced draft, kilowatts	32	30	29	29	31	29	28	27	40	36	34	31	33	30	28	27
For circulating water, kilowatts	188	171	158	157	177	160	147	143	178	155	138	132	178	155	138	130
For dry vacuum, kilowatts	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17
Total auxiliary power consumed, kilowatts	475	531	687	874	440	483	604	746	497	559	746	1006	457	501	631	793

TABLE 7 CALCULATIONS FOR ESTIMATED BOILER-ROOM HEATING SURFACES REQUIRED FOR 30,000 KW. GROSS TURBO-GENERATOR OUTPUT

(Throttle temperature, 700 deg. fahr.)

	Cycle A (Rankine)				Cycle B (Reheating) (For reheating pressure = 30 per cent of throttle pressure)				Cycle D (Regenerative)				Cycle E (Reheating-Regenerative) (For reheating pressure = 40 percent of throttle pressure)			
Throttle pressure, lb. per sq. in. abs. . .	200	400	800	1200	200	400	800	1200	200	400	800	1200	200	400	800	1200
Ideal Cycles:																
Feed temperature leaving heaters, deg. fahr.	79	79	79	79	79	79	79	79	235	299	386	457	171	218	276	319
Heat supplied to steam, B.t.u. per lb.:																
(a) By boiler, superheater and economizer	1327	1315	1286	1251	1327	1315	1286	1251	1171	1095	974	863	1235	1177	1088	1010
(b) By reheater	0	0	0	0	140	146	162	182	0	0	0	0	111	116	131	149
(c) By total heating surface	1327	1315	1286	1251	1467	1461	1448	1433	1171	1095	974	863	1346	1293	1219	1159
Real Plant:																
Estimated boiler pressure, lb. per sq. in. abs.	210	417	841	1263	208	412	819	1226	211	420	836	1256	209	414	823	1232
Total steam flow, lb. per hr.	293,000	276,000	268,000	273,000	257,500	240,000	229,000	228,000	308,000	298,000	310,000	339,000	273,000	257,000	250,000	254,000
Estimated feed temperature to economizer, deg. fahr.	75	75	75	75	75	75	75	75	188	252	344	423	121	167	225	268
Heat supplied to steam, B.t.u. per lb.	1331	1319	1290	1255	1449	1443	1431	1418	1218	1142	1018	900	1378	1326	1255	1195
Millions of B.t.u. supplied per hr.:																
(a) To the steam	390	364	346	343	373	346	328	323	375	341	316	305	376	341	314	304
(b) By the coal	464	433	411	408	444	412	390	385	446	406	376	363	448	406	374	362
Coal[1] required in lb. per hr.	37,700	35,200	33,500	33,200	36,100	33,500	31,700	31,400	36,300	33,100	30,600	29,500	36,500	33,100	30,400	29,500
Heating surfaces to give 84 per cent efficiency:																
(a) Boiler	57,000	52,400	49,000	48,400	49,600	45,000	41,200	39,400	55,200	49,600	45,800	44,600	50,500	44,400	40,200	37,900
(b) Superheater	13,900	11,900	10,100	8850	12,200	10,400	8,650	7,400	14,500	12,900	11,700	11,000	12,900	11,100	9,450	8,250
(c) Reheater	0	0	0	0	8,200	8,050	8,700	10,000	0	0	0	0	6,850	6,900	7,800	9,200
(d) Economizer	25,400	27,900	30,900	32,900	24,800	27,100	29,800	31,900	26,300	29,800	32,800	36,500	30,200	40,800	35,600	36,300
(e) Air heater	0	0	0	0	0	0	0	0	11,400	18,000	28,500	37,500	0	0	11,500	17,700
Total	96,300	92,200	90,000	90,150	94,800	90,550	88,350	88,700	107,400	110,300	118,800	129,600	100,450	103,200	104,550	109,350

[1] Heating value taken as 12,300 B.t.u. per lb.

as the pressure is increased, and how this power goes up with the increased flow of water which is necessary with the regenerative cycles D and E, as compared with those in which there is no bleeding. In Fig. 23 may be seen the curves which show the power required by all of the auxiliaries for each of the four cycles and all pressures from 200 to 1200 lb. per sq. in.

Heating Surfaces

59 For the purpose of this study it was believed that the best comparative results concerning the fuel consumption to be ex-

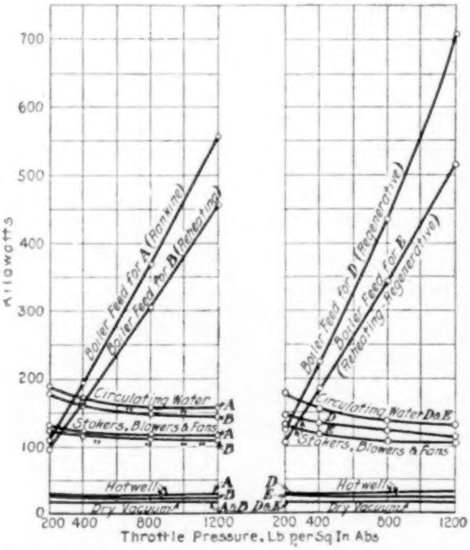

Fig. 22 Estimated Auxiliary Power for Boiler and Condenser Rooms

pected in plants of the same general type, modified for the different pressures and cycles, would be obtained by keeping the boiler efficiency the same for all cases. A figure of 84 per cent was therefore chosen as being a fair value to use when the plants are running at 100 per cent load with the boilers operating at 160 per cent rating. The coal was assumed to contain 12,300 B.t.u. per lb. as fired and 13.65 lb. of dry gas were assumed for each pound of coal as fired. The gases leaving the boiler were assumed to be maintained at a temperature 140 deg. higher than the temperature of saturated steam corresponding to the boiler pressure. The heat transfer in the superheater and the reheater was assumed to be 3700 B.t.u. per hr. per sq. ft. of surface for all the cases considered. For the economizer the heat transfer was assumed to be at the rate

of 5 B.t.u. per hr. per sq. ft. per deg. fahr. mean temperature difference. For the air preheater the heat-transfer rate was taken as 60 per cent of the rate used for the economizer, or 3 B.t.u. per sq. ft. per hr. per deg. fahr. mean temperature difference. With these assumptions the heating surfaces necessary to give 84 per cent efficiency at 160 per cent of rating are given in Table 7.

60 The steam flow for the different cycles and the corresponding temperatures of the feedwater to the economizer are shown by the curves of Fig. 24. Mention should be made of the fact that for cycles A and B the feedwater should be heated to a higher temperature than 75 deg. before being fed to the economizer in order

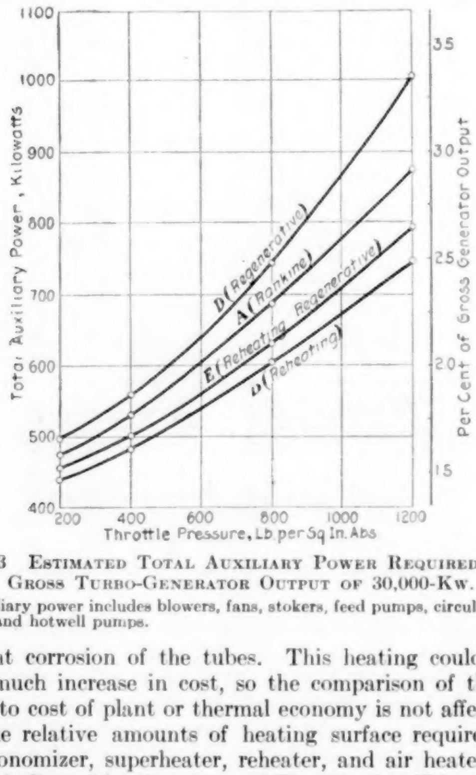

FIG. 23 ESTIMATED TOTAL AUXILIARY POWER REQUIRED FOR A GROSS TURBO-GENERATOR OUTPUT OF 30,000-KW.

This auxiliary power includes blowers, fans, stokers, feed pumps, circulating pumps, air pumps, and hotwell pumps.

to prevent corrosion of the tubes. This heating could be done without much increase in cost, so the comparison of the several cycles as to cost of plant or thermal economy is not affected.

61 The relative amounts of heating surface required for the boiler, economizer, superheater, reheater, and air heater are perhaps best indicated by the curves of Fig. 25. It will be noted that for the high pressures, both of the regenerative cycles require an air preheater in order to give an efficiency of 84 per cent. This is due, of course, to the high feedwater temperature as shown by the curves D and E of Fig. 24. For pressures less than 800 lb. per sq. in. the use of preheaters is not so essential with the reheating-regenerative cycle E as it is with D, which has no reheating.

Station Piping

62 In selecting the proper size of steam mains to be used with the high pressures there is no very definite information available at the present time as to what steam velocity is best. There is also a great difference of opinion as to what constitutes the proper length of steam line from boilers to turbine in making an estimate of piping cost.

63 The estimates given in this paper were based upon a general

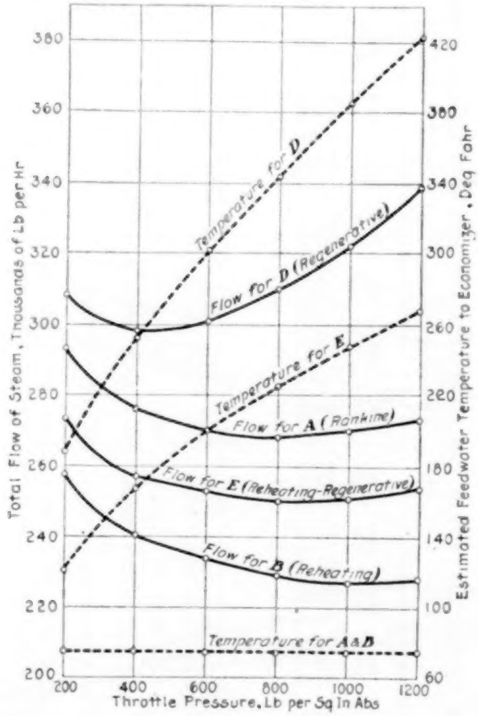

Fig. 24 Estimated Steam Flow and Feedwater Temperature as Determined by Various Pressures and Cycles for 30,000-Kw. Gross Turbo-Generator Output

arrangement which was laid down so that each turbine was normally supplied with steam from three boilers, suitable cross-connections being provided to take care of emergencies. Special effort was made to keep the general piping arrangement the same for each of the plants except as modified for the ones using reheating. For the cycles (A and D) which have no reheating the length of steam line from boiler to turbine was taken as 200 ft. of main with 100 ft. of smaller size for the connections to the three boilers. For the

TABLE 8 ESTIMATED COST OF STATION PIPING FOR THREE BOILERS AND ONE 30,000-KW. TURBO-GENERATOR

	Cycle A (Rankine)						Cycle B (Reheating) and Cycle E* (Reheating-Regenerative)						Cycle D (Regenerative)					
Throttle pressure, lb. per sq.in. abs.	200	400	600	800	1000	1200	200	400	600	800	1000	1200	200	400	600	800	1000	1200
Throttle flow, thousand lb. per hr.	293	276	270	268	270	273	257.5	240	234	229	227	228	308	298	301	310	322	339
Main steam line, outside diam., in.	16.00	11.75	9.625	8.625	8.625	7.625	15.00	10.75	9.625	8.625	7.625	7.625	16.00	11.75	10.75	9.625	8.625	8.625
Main steam line, inside diam., in.	15.25	11.00	8.725	7.585	7.325	6.305	14.25	10.03	8.725	7.565	6.425	6.305	15.25	11.00	9.75	8.425	7.325	7.125
Main steam line, wall thickness, in.	0.375	0.375	0.450	0.520	0.650	0.660	0.375	0.360	0.450	0.530	0.580	0.660	0.375	0.375	0.500	0.600	0.650	0.750
Velocity, thousand ft. per min....	13.2	11.5	11.6	11.4	9.2	10.1	13.0	11.9	9.8	9.9	10.7	8.4	13.8	12.6	10.6	10.8	11.1	9.6
Cost of piping, erected and covered, in thousands of dollars:																		
(a) Main steam (inc. reheating)	35.9	32.8	34.5	34.3	37.3	38.0	71.1	61.6	66.9	64.1	68.7	73.5	35.9	32.8	36.5	37.7	37.3	40.8
(b) Boiler feed................	26.6	35.3	41.6	50.1	56.2	61.7	26.6	35.3	40.0	47.1	52.5	57.6	29.2	40.8	51.1	59.8	66.2	73.2
(c) Blow off..................	11.2	13.0	14.0	15.8	17.0	17.7	11.2	13.0	14.0	15.8	17.0	17.7	11.2	13.0	14.0	15.8	17.0	17.7
(d) High-pressure drip and instruments..............	6.4	8.0	9.2	10.8	11.9	13.0	6.4	8.0	9.2	10.8	11.9	13.0	6.4	8.0	9.2	10.8	11.9	13.0
Total for (a), (b), (c), (d)....	80.1	89.1	99.3	111.0	122.4	130.4	115.3	117.9	130.1	137.8	150.1	161.8	82.7	94.6	110.8	124.1	132.4	144.7
Other station piping, not affected by steam pressure.......	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1	148.1
Total cost of all piping, thousands of dollars..........	228.2	237.2	247.4	259.1	270.5	278.5	263.4	266.0	278.2	285.9	298.2	309.9	230.8	242.7	258.9	272.2	280.5	292.8

* The values for Cycle E are not strictly the same as for Cycle B, since the steam flow for E is from 6 to 10 per cent greater (See Table 5)

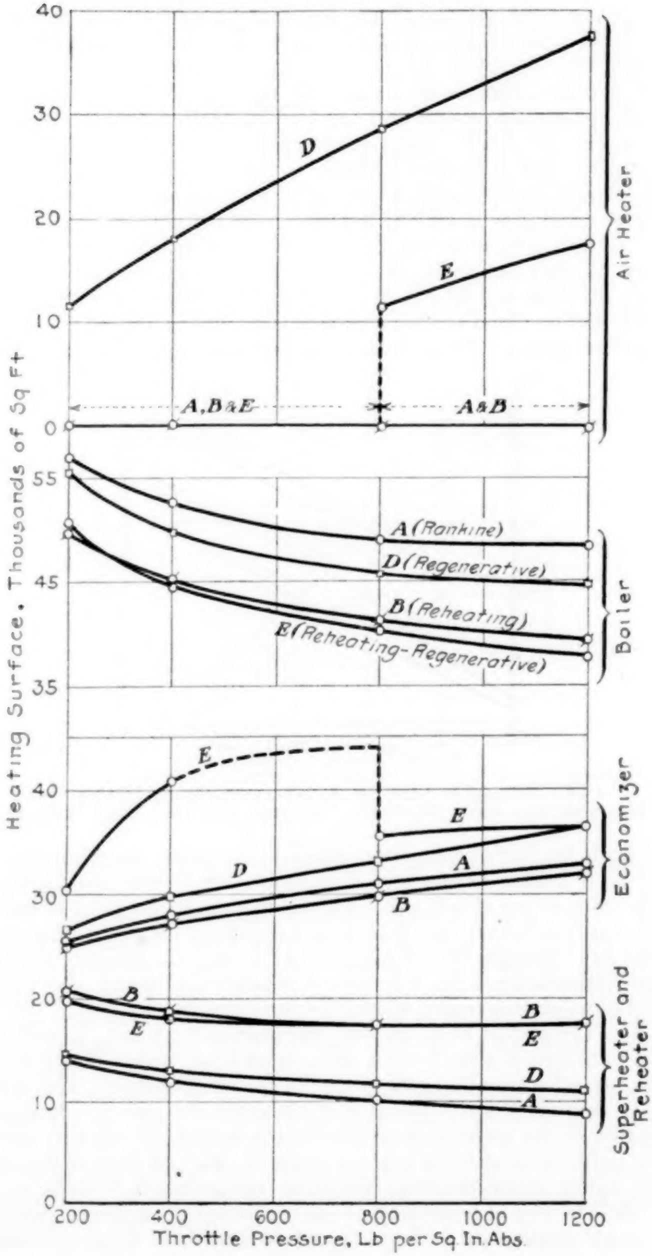

Fig. 25 Heating Surfaces Required to Give an Efficiency of 84 Per Cent with Steam Flow and Feedwater Temperature as Shown by Fig. 24

reheating cycles (B and E) the length of piping from the boiler to the turbine was taken as 150 ft., which is possibly too small, and 450 ft. of a larger size for the reheating. In Table 8 will be found the detailed information concerning the size of pipe, wall thickness, and the corresponding steam velocities selected for the various pressures and cycles under consideration. It will be noted that standard sizes of outside diameter of the pipe have been adhered to. These sizes naturally give slightly illogical variations of steam velocity in a few cases. It is believed by the authors that as we go to the higher pressures the steam velocity should be somewhat reduced, and the values given in Table 8 will indicate how this idea has been carried out in this study. In the lower half of the table will be found the detailed estimates for the cost of the high-pressure piping. These estimates must naturally be considered

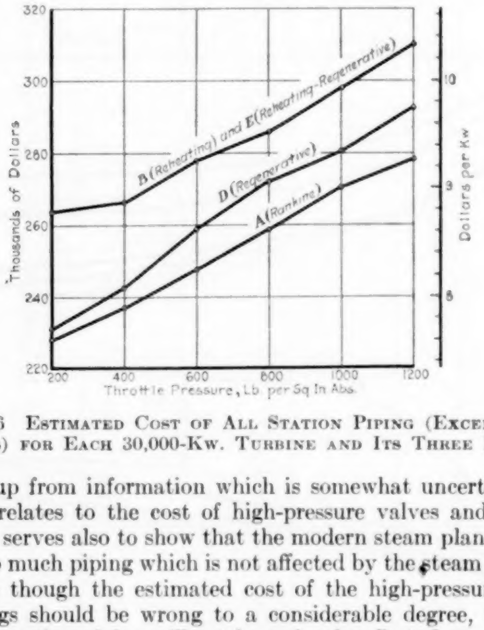

Fig. 26 Estimated Cost of All Station Piping (Except for Heaters) for Each 30,000-Kw. Turbine and Its Three Boilers

as made up from information which is somewhat uncertain in so far as it relates to the cost of high-pressure valves and fittings. The table serves also to show that the modern steam plant of large size has so much piping which is not affected by the steam pressure, that even though the estimated cost of the high-pressure valves and fittings should be wrong to a considerable degree, the total cost of all station piping will not be seriously affected.

64 The cost of bolts, gaskets, hangers, anchors, etc., as well as the total cost of labor for the main steam lines, was assumed to be the same for equal lengths of piping for all pressures. This was done because it was believed that, although the pipe sizes became smaller as the pressures increased, the increased difficulty in making tight joints and successfully hanging, guiding, and anchoring the piping under increased pressures, compensated for this fact.

65 The cost of boiler-feed piping was based upon data obtained from an actual plant having a flow of 300,000 lb. of water per hour.

For each of the plants here considered the cost was altered to agree with the flow expected and also to take care of the increased pressure. The cost of labor was assumed to increase at the rate of 5 per cent for each 200-lb.-per-sq.-in. rise in pressure. It may be noted from Table 8 that the cost of the feedwater piping increased rather rapidly with the pressure. This is due to the fact that the pipe sizes remain nearly constant regardless of the pressure involved. All boiler-feed pumps were considered as motor-driven and no piping was figured for a house alternator, or in other words, the estimate does not include any steam piping for superheated steam to auxiliaries. This statement is not to be construed to mean that the authors believe that boiler-feed pumps should always be driven electrically, or house alternators eliminated. This method

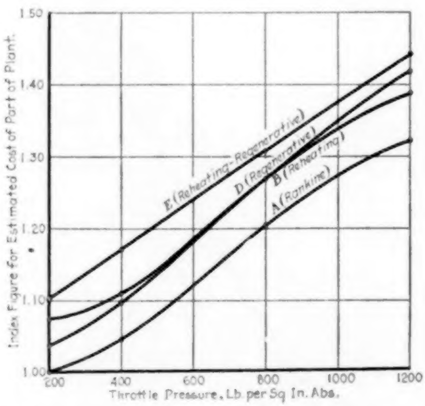

FIG. 27 ESTIMATED RELATIVE COSTS OF CENTRAL-STATION APPARATUS AFFECTED BY STEAM PRESSURE AND CYCLE

was chosen as being the simplest and fairest one for the purpose of this study.

66 The cost of the portion of the blow-off piping directly exposed to boiler pressure was estimated to vary with the pressure in the same ratio as the cost of the boiler-feed piping, and the basic unit costs were obtained from an actual plant. The cost of the section of blow-off piping between the last valve in the line and the atmospheric discharge was taken to be the same for all pressures.

67 In the estimate of the cost of high-pressure drip and instrument piping, the unit costs were again obtained from an actual plant and increased with the pressure in the same ratio as the cost of boiler-feed piping. In estimating the total cost of all piping not affected by pressure, the values were taken the same for each pressure in each cycle.

First Cost of Plant

68 The cost of all station piping except that required for the extraction heaters has been discussed in the preceding section. The cost of the piping for the extraction heaters, which is less than 3 per cent of the total station piping, has been included with the condenser cost, which was taken as proportional to the surface already given in Fig. 21.

69 The cost of all equipment which is affected by the steam

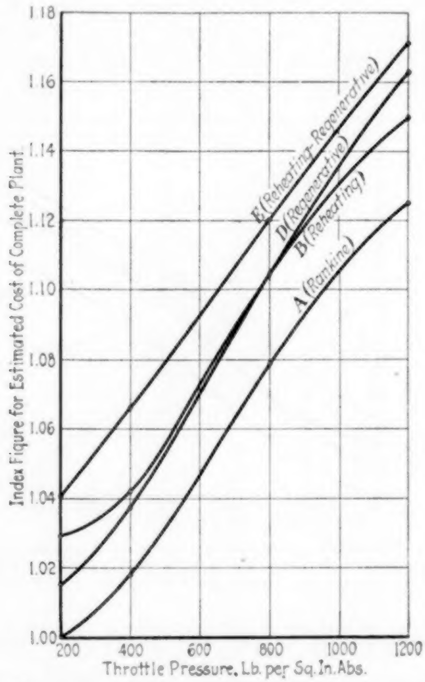

Fig. 28 Estimated Relative Costs of Complete Central Stations, Including Buildings and Land

pressure was determined and the corresponding index figures are given by the curves of Fig. 27. From these curves it may be noted that increasing the pressure from 200 to 1200 lb. per sq. in. will increase the cost of this equipment about 30 per cent for any particular cycle.

70 For the cost of auxiliary apparatus, the estimate was based upon motors large enough to operate the boilers at 300 per cent of rating. The circulating pumps were supplied with motors having a normal rating 20 per cent greater than the power calculated in Table 6.

71 The cost of the air heater was taken as $1.60 per sq. ft. of heating surface. This figure is intended to cover the cost of flues and all other expenses connected with the installation.

72 If to the cost of equipment affected by steam pressure there is now added the cost of land, canals, railroads, foundations, buildings, coal handling, miscellaneous mechanical equipment, and all electrical equipment except the generator, new index figures are

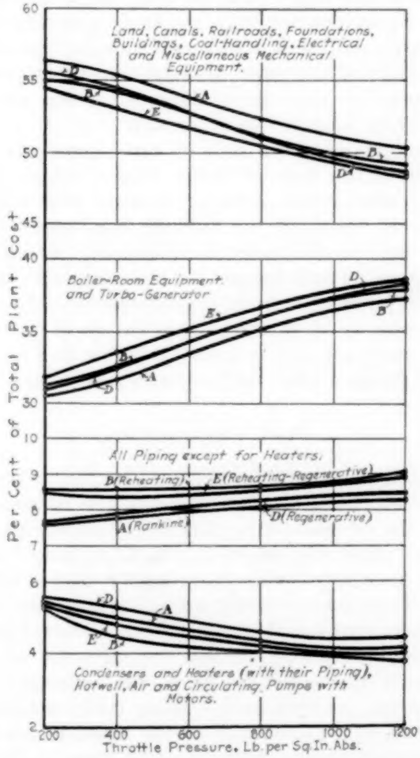

FIG. 29 ANALYSES OF PLANT INVESTMENT

obtained as shown by Fig. 28. From these curves it would appear that the 1200-lb. plant will cost only about 14 per cent more than the 200-lb. one to operate on the same cycle. For many cases this increase in cost will be greater when the higher pressures are used, for the reason that some plants will have a higher portion of the total plant cost dependent upon pressure than was used in this study. It is also clear that the plant built to operate upon the Rankine cycle is appreciably lower in first cost than any of the others, and the reheating-regenerative plant is the most expensive throughout the entire range of pressures considered.

73 The cost of land, buildings, etc., was taken the same for each plant considered, but, due to the increase in the cost of equipment affected by pressure, this represents a decreasing percentage of the total plant investment as shown by the curves at the top of Fig. 29. These curves also show that in going from a pressure of 200 to 1200 lb. the investment in land, buildings, etc., varies from about 57 to 49 per cent of the total for nearly all of the cycles. Other curves of this figure indicate that, for pressures from 200 to 1200 lb. per sq. in., respectively, the turbo-generators and boiler-room equipment, with feed pumps and motors, require from about 31 to 38 per cent of the entire investment; the piping costs from about 7 to 9 per cent, and the condensers, heaters, and pumps with motors cost from about 5 to 4 per cent.

74 The variation in the cost of turbo-generators and boilers is given only in the form of index figures, which represent the best average values which could be obtained from the information originally in the hands of the authors, combined with that procurable from the manufacturers of equipment. Few manufacturers have actually developed designs for the very high-pressure apparatus, and still fewer have built such equipment. As a result there is much uncertainty as to the proper values of the index figures for the high pressures, and it is also probable that they now carry development charges which will be amortized later.

Fuel Consumption

75 The coal consumption of plants operating under different conditions as to steam pressure and cycle, is of great interest to engineers at the present time. For the pressures and cycles considered in this study the curves in Fig. 30 represent unbiased estimates which have been made in a manner that is believed to be reasonable. Time did not permit preparing diagrams in which loads less than 100 per cent would be considered. The general relation of the curves would, however, remain the same for all cases except those in which the stand-by losses are excessive. The term "capacity factor" as used in this study really means the station capacity factor, which is defined as the ratio of the gross station output in kilowatt-hours during a given time to the station capacity in kilowatts multiplied by the number of hours in the period considered.

76 From the curves in Fig. 30 it is easy to estimate the amount of money that may be saved in fuel by one type of plant relative to another in any given time provided the quality and cost of the fuel are known. For the pressures between 200 lb. and 400 lb. the plant operating on the Rankine cycle requires about 5 per cent more fuel than the others and this inferiority is worse at higher temperatures. For the higher pressures the application of the regenerative principle will save fuel, as may be clearly seen from the comparison of curves D and E with A and B. Attention

is called to the fact that the curves of Fig. 30 tie back to those of Fig. 20, which shows the engine efficiencies, and that in deriving the latter it was not assumed that any particular provision was made for extraction of water formed during expansion in the turbine. If it be assumed possible to extract such water, as, for example, by drainage to the heaters, the efficiency of the turbine

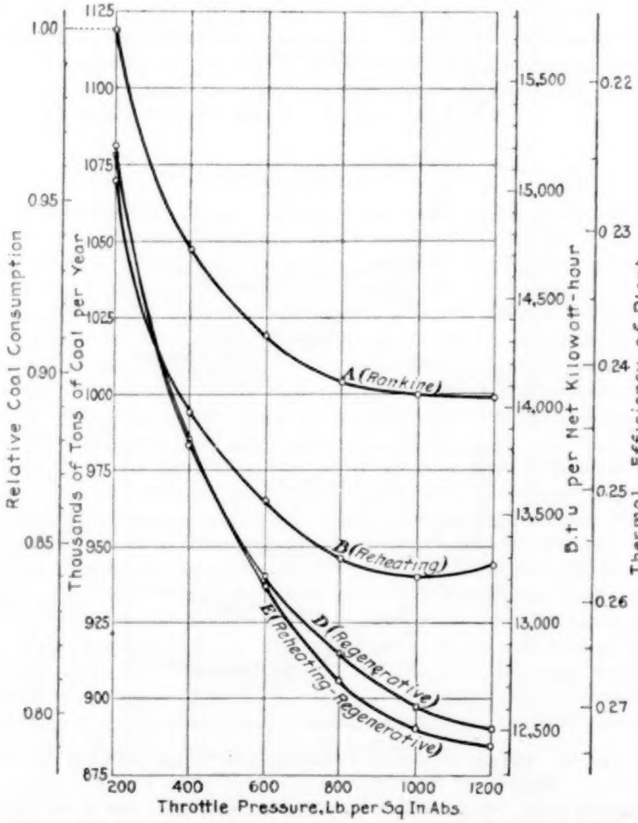

FIG. 30 RELATIVE COAL CONSUMPTION OF 200,000-KW. PLANTS OPERATING ON VARIOUS CYCLES AND STEAM PRESSURES

Capacity factor, 100 per cent; boiler efficiency, 84 per cent; throttle temperature, 700 deg. fahr ; exhaust pressure, 1 in. Hg abs.; heating value of coal, 12,300 B.t.u. per lb.

operating on the regenerative cycle will be improved and the fuel consumption of the plant will be less than that shown in Fig. 30.

77 For pressures of less than 400 lb. per sq. in. it appears, from Fig. 30, to make but little difference what cycle other than the Rankine is chosen from the consideration of fuel alone.

78 For the conditions given in Fig. 30, it may be seen that a plant operating on the reheating-regenerative cycle at a pressure of 1200 lb. would only require about 79 per cent as much fuel as the one operating on the Rankine cycle at 200 lb., and for these same pressures the plant using the regenerative cycle would require only about 80 per cent as much fuel as the Rankine. In both

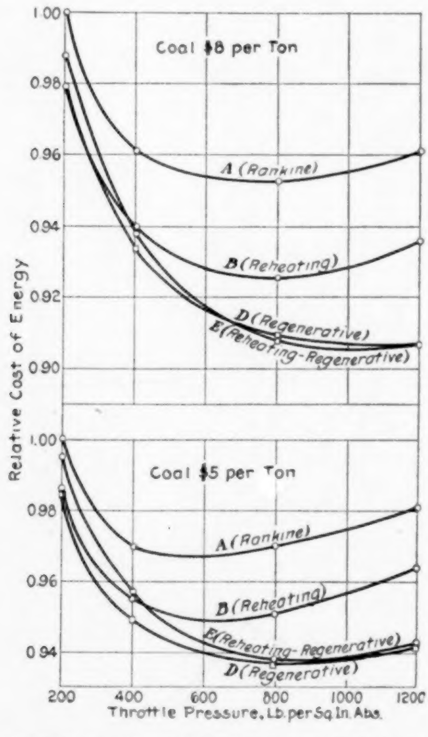

FIG. 31 EFFECT OF STEAM PRESSURE AND CYCLE ON COST OF ENERGY AT SWITCHBOARD OF 200,000-KW. PLANT

Capacity factor, 100 per cent; boiler efficiency, 84 per cent; cost of coal, $5 or $8 per ton as indicated; cost of energy taken as 1.00 for the Rankine cycle at 200 lb. per sq. in. abs.

cases no provision for water extraction has been assumed. These are important ratios requiring serious consideration for plants with loads that can justify the extra first cost that goes with such high pressure.

79 It is worth while to note the striking similarity between the results indicated in Figs. 18 and 30. Comparison of the values shown in these two figures gives a ready means of determining the extent to which the relative fuel economy of real plants is affected

by the variation of the real machines from the ideal and the necessary use of auxiliary power, as both of these factors vary with different cycles. The plant with the reheating cycle is better than any of the others in both of these respects, i.e., it most nearly approaches its ideal, as may be seen from Figs. 20 and 23, but this superiority is not sufficient to overcome the better efficiency of the ideal regenerative cycles, shown by Fig. 18, except at pressures less than about 300 lb. per sq. in. and on the assumption that water is not drained during expansion. Above this pressure the great superiority of the ideal cycles involving the regenerative principle overcomes the extra energy losses encountered when trying to apply this principle in the real plant.

Relative Costs of Energy

80 The amount of fuel burned to generate a unit of electrical energy at the switchboard of a modern central station is a large item in the total cost of this energy, but it is far from all, as the operating maintenance and fixed charges are very large. The curves of Fig. 31 show the relative costs of the energy delivered at the switchboard of the various types of plants under consideration in this study. The curves have been drawn so that the index figures all refer to the Rankine cycle with a pressure of 200 lb. per sq. in. as the standard of comparison. The throttle temperature is assumed to be 700 deg. fahr. in all cases.

81 From the curves of Fig. 31 it appears that with coal at $5 per ton the regenerative cycle is superior to the others for all pressures, but compared with the reheating-regenerative this superiority is very small for pressures above 800 lb. per sq. in. It is also clear from these curves that with five-dollar coal, pressures higher than 600 lb. are not so attractive as sometimes believed. As the capacity factor is decreased to values less than 100 per cent the high pressures are still less attractive. With coal costing $8 per ton and with base-load conditions the best pressure would appear to be near 1000 lb. per sq. in., and for these conditions the regenerative and the reheating-regenerative are the best cycles, except for pressures less than 400 lb. per sq. in.

Turbine Room Economy

82 The curves shown in Fig. 32 have been drawn for those primarily interested in the economy of the equipment in the turbine room as contrasted with the plant as a whole. This figure shows the effect of steam pressure and cycle on the thermal efficiency of the turbo-generator, the heat supplied to the turbine being measured above the temperature of the feedwater to the economizer in each case. This thermal efficiency is equal to the ideal cycle efficiency from Fig. 18, multiplied by the engine efficiency, from Fig. 20. Fig. 32 shows a thermal efficiency very

close to 34 per cent at 1200 lb. per sq. in. for both the regenerative and the reheating cycles, the corresponding heat supplied to the turbine by the steam generating equipment being in the neighborhood of 10,000 B.t.u. per kw. hr.

83 While it is true that the turbine alone is not chargeable with all of the losses, as given by Tables 3 and 4, it is most con-

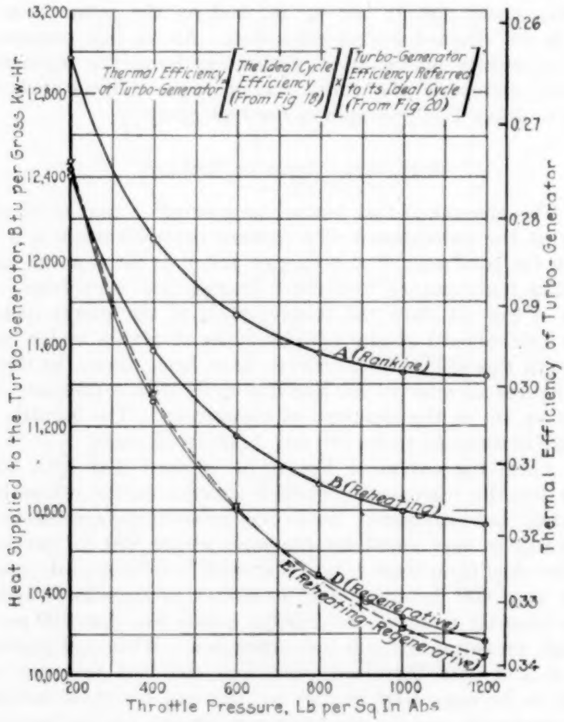

FIG. 32 EFFECT OF STEAM PRESSURE AND CYCLE ON THE THERMAL ECONOMY OF THE TURBO-GENERATOR

Throttle temperature, 700 deg. fahr.; exhaust pressure, 1 in. Hg abs.; generator efficiency, 97 per cent.

venient in a study of this kind as in many other cases, to use one efficiency to express the direct relation between the efficiency of the ideal cycle and thermal efficiency of the real turbo-generator group, with its heat supply measured above the temperature of the feedwater to the economizer. The same sort of question is involved in the discussion as to whether the supply to a turbine operating on the Rankine cycle should be measured above the temperature in the hot well or above feedwater temperature corresponding to the pressure in the exhaust nozzle. For the

Rankine cycle this difference is not large but in case of the regenerative cycles it is very appreciable. The comparison will be fair for the different plants so long as one method is followed throughout the discussion.

Operating Characteristics

84 For all plants there is some definite limit of initial expenditure and degree of complication beyond which one is not justified in going for the purpose of obtaining fuel economy. This limit varies with local conditions to such an extent that no fixed and universal rule can be predicated. However, it is almost axiomatic that that plant which will achieve a certain thermal efficiency with the minimum initial expenditure, minimum complication, and minimum danger of breakdown or disarrangement of parts is the most desirable under any conditions.

85 Experience with high-pressure equipment and with auxiliary apparatus which the use of high-pressure steam appears to involve is as yet exceedingly limited. No one is today really in position to say positively that the high-pressure equipment now coming into use will operate with that smoothness and certainty required to justify its use for the sake of the increased thermal efficiency obtainable. Under such conditions it would seem to be best to use first the simplest types and to progress toward the more complicated if further experience indicates such progress to be desirable. From such a point of view the simple Rankine cycle would be best if it gave sufficient promise of increased efficiency with increased pressure. Unfortunately it does not, and it is therefore necessary to consider a more complicated cycle of operation. Our present knowledge indicates two choices, that involving regenerative heating only and that involving reheating with or without regeneration.

86 Of the two, the regenerative cycle is unquestionably the simpler, and in fact it is hardly more complicated than the Rankine now in use. All additional equipment involved is of well-tried types, is operated at comparatively low temperature and probably at low pressure, and it introduces few operating complications. No additional functions are required of the governor, this mechanism operating exactly as it would with a simple Rankine cycle.

87 Attention is called to the fact that it is necessary to provide against runaway through vaporization of hot water contained in stage heaters. Several methods of doing this have been evolved and no great degree of complication is introduced.

88 In contrast to this, the reheating cycle involves the use of a reheater with connecting piping and provision against runaway caused by the large steam content of the reheating system. As developed to date, this requires additional features in the governor mechanism and therefore additional complications. There do not

appear to be any insurmountable difficulties, but it is certain that operation will be more complicated for an equal degree of reliability.

89 It seems probable that the reheater will in most cases be located in the boiler or boilers, involving the return of medium-pressure or low-pressure steam to the boiler room and back again to the turbine. When the difficulty experienced in obtaining a satisfactory disposition of high-pressure steam piping in a modern station is considered, it does not appear that two additional pipes of larger diameter can be included without radical modification of design or of our ideas with regard to desirable clearances, ease of access, etc.

90 In this connection it is interesting to note that one possible solution of this problem which would seem to reduce the complications has been suggested. This suggestion is based on the assumption that little if any difficulty would be experienced in superheating steam initially to a temperature about one hundred degrees higher than the value desired at the turbine throttle. It is therefore proposed to pass such highly superheated steam through a reheater located at the turbine, utilizing its excess superheat to reheat steam withdrawn from the turbine. A reheater of this type would certainly be large because of the comparatively low temperature head and the low transmission coefficient to be expected, but its use would unquestionably simplify the piping and probably the governor problem.

91 With the highest pressures discussed it would appear that the reheating feature should be combined with the regenerative to obtain an efficiency better than the simple regenerative, so that the combination must without doubt be more complicated both in design and operation than the simple regenerative system.

92 The use of superheated steam in the low-pressure part of the turbine also adds complications. Steel casings must be used, thus adding to the cost. High-temperature and variable-temperature problems are introduced at two points instead of at one point in each unit.

93 Viewing the reheating in this way it would seem that efforts directed toward the perfection of equipment for getting the most out of the regenerative cycle promise the largest return in early development, low cost, operating simplicity, and thermal economy.

Possibilities and Probabilities

94 Considering all of the factors which enter into the problem, is seems to the authors that the high-pressure regenerative plant is the most promising for commercial development. It is certainly true that the performance here estimated for such a plant can be further improved by using a turbine designed to separate and remove water formed during expansion. Some of the designs now available provide for such drainage at the bleeder points to a

certain extent, but it seems probable that the best results cannot be attained unless effective moisture separators are actually built into the turbine structure.

95 Granting, for the sake of argument, that such machines can be built, we should have available a plant capable of giving better economy than indicated for the reheating-regenerative type. Moreover it would have certain additional advantages in comparison with the latter. The quantity of steam passing through the low-pressure stages for a given electrical output would be less and the condenser smaller. The complications in design and operation resulting from the necessity of using reheaters would be avoided also.

96 The cost of stage heaters and their connections has been the subject of adverse comment by several engineers who have considered regenerative designs. This cost standing by itself looks large, but when credit is taken for the decreased size of the condenser the situation does not look so serious. In effect the stage heater is merely condenser surface which has been removed from its customary position. Moreover the use of such heaters makes it possible to modify the turbine design so as to use a smaller low-pressure end with a given high-pressure end, or, a larger high-pressure end with a given low-pressure end, thus obtaining obvious additional advantages.

97 Further, these stage heaters have thus far been constructed only in the surface type, that is, as surface heaters or condensers. There is the possibility of further reduction in cost by the use of contact heaters of some sort, for example, jet condensers, eductor condensers, and the like. It seems probable that such devices might be used at the higher-pressure points, and particularly if bleeding is carried into the superheated field.

98 It is essential to note that economizers become of less value with the regenerative cycle as the initial steam pressure is increased. This naturally follows from the increasing temperature of feedwater leaving the regenerative heaters. The high turbine-room efficiency is therefore obtained to a certain extent at the expense of boiler-room possibilities, and this suggests immediately that air heaters be used with such plant to conserve waste heat not available for use in economizers fed with high-temperature water.

99 It is obvious that it will be necessary to heat the air in such devices to temperatures from 100 to 200 deg. fahr. above normal air temperature if full conservation of waste heat is to be made. The effect of such high air temperatures on stokers, arches, furnace walls, and slagging of tubes is practically unknown, but the authors believe that no insurmountable difficulties will be presented. This belief is based upon fragmentary experiences which, when taken together, make a fairly strong case for the successful use of the air heater under the conditions that are necessary to obtain the best results with high-pressure regenerative cycles.

Acknowledgments

100 The authors desire to express their thanks to Mr. W. H. Patchell of London, England, and to Messrs. C. H. Berry, P. W. Thompson, and A. K. Bak, engineers with The Detroit Edison Company, for their material assistance in the collection of technical information and in the development of the discussion in the various fields in which they are especially qualified. In particular they desire to thank Mr. F. A. Kohlmeyer for estimating the cost of the station piping, Mr. R. B. Purdy and Mr. C. H. Best for calculating the various heating surfaces and auxiliary power required, Mr. S. S. Sanford for editing the text, and Mr. K. P. Kammer for making the drawings.

General Conclusions

101 The studies outlined in this paper, together with others with which the authors are familiar, indicate plainly that the improvement in economic results to be expected from the use of higher steam pressures in plants designed to take full advantage of the possibilities latent in the use of such pressures are sufficiently great to make it appear quite probable that the more progressive engineers and executives will construct plants of this character in ever-increasing numbers.

102 It must be recognized that high-pressure equipment necessarily carries high development charges at the present time, and that this fact, coupled with uncertainty with respect to the performance of equipment of untried types, must to a certain extent retard its adoption. It is therefore to be expected that higher pressures will be adopted first and more frequently in connection with plants of the base-load type and in regions in which fuel costs are high.

103 The authors feel that steam pressures of the order discussed in this paper should no longer be regarded as of theoretical interest only. Most of the major problems involved in the design and arrangement of equipment for utilizing such pressures have been solved or are nearing what appear to be satisfactory solutions, and it is believed that careful engineers who are thoroughly familiar with the peculiar features involved in this sort of work can safely install equipment for even the highest pressures here considered when the circumstances and conditions of use justify such installations.

104 Certain of the factors in this paper have been calculated for maximum temperatures of both 700 and 800 deg. fahr. The authors feel that calculations for 800 deg. fahr. are at the present time of minor interest, as they do not believe that materials and designs now available may be considered adequately safe for use with such temperatures. It is exceedingly difficult to state categorically any definite upper limit of temperature, but it is felt that

a figure between 700 and 750 deg. fahr. represents the safest upper limit of steam temperature at the present time.

105 The authors believe that this study indicates that even with the present type of turbine the regenerative cycle is the best one to use in large stations, as it ranks very high from the standpoint of fuel consumption, operating characteristics, and first cost throughout the entire range of pressures, but particularly so at the higher ones. With a turbine which the authors believe can be developed so as to remove the moisture to a considerable degree, this cycle will give still better economy, and it seems altogether reasonable to expect that this equipment in the turbine room, combined with that now developed to yield high boiler-room efficiency, will give station economy that will pay handsomely for the increased investment.

NOTE REGARDING APPENDICES

Appendix No. 1, comprising Tables 9 to 17, inclusive, which present detailed calculations for the various cycles, and Appendix No. 2, a bibliography of material published since 1917, are not published in this volume. Copies are on file at the Society headquarters and will be loaned upon request.

No. 1913c

ECONOMY CHARACTERISTICS OF STAGE FEEDWATER HEATING BY EXTRACTION

PRACTICAL COMPUTATION AND MAXIMUM REALIZATION OF EXTRACTION GAINS FOR POWER PLANTS OF PRESENT TREND IN DESIGN

By E. H. Brown,[2] Milwaukee, Wis.
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Non-Member

Feedwater heating by extraction from the main unit, becoming increasingly important with rising steam pressures, complicates the computation of heat balance. The authors point out the influence of each possible factor in the computations, present a practical method for computing extraction cycles, and deal with various factors affecting turbine-room economy, these being more particularly enumerated in the first three paragraphs of the paper.

WHEN one realizes that comptometers are displacing slide rules for power-plant calculations, that nearly every heat unit is being accounted for, and that resultant power units are no less immune in the ever-growing crusade for maximum economy, then he appreciates that present power-plant requirements demand the nearly perfect agreement between theory and practice which unfortunately has been so long in attainment.

2 Feedwater heating by extraction from the main unit, becoming increasingly important for both thermodynamic and practical reasons with rising steam pressures, complicates the computation of heat balance. An error of 0.5 per cent in overall plant economy may easily be occasioned by a single incorrect assumption or omission due to lack of understanding in the extraction characteristics of steam turbines. To point out the influence of each possible variable in the computations and finally

[1] For discussion and closure see pp. 766 and 812.
[2] Steam Turbine Dept., Allis-Chalmers Mfg. Co.

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to illustrate a comprehensive though not tedious method for their allowance, is the first purpose of this paper.

3 Where heaters can be most efficiently placed thermodynamically, how their number and the temperature to which they work influence turbine-room economy, to what extent auxiliary equipment limits the gains acquired by extraction heating, how the type of turbine employed affects the same benefits, and how other factors affect turbine-room economy, were ascertained by accurate computations using the method mentioned, taking into account all conditions as they actually exist in representative installations, and are submitted in the second portion of the discussion. Turbine-room variables and approximations alone have been attacked, and it is hoped that boiler-room computations may be similarly approached so that the average power-plant designer or operator, aided only by manufacturers' guarantees and an elementary knowledge of thermodynamics, may accurately compute a representative heat balance for any installation.

I—PRACTICAL COMPUTATION OF EXTRACTION CYCLES

4 Neglecting generator and mechanical losses when computing heat consumption in extraction feed-heating systems may result in an error of 0.5 per cent in overall turbine-room economy; ignoring the change in stage efficiencies may amount to nearly the same error; or assuming no terminal difference at the heaters nor moderate pressure drops to them or at the extraction nozzles may cause errors of 0.6 per cent and 0.1 per cent each, respectively. Taking into account flashing of the drained condensate from heater to heater, as actually occurs, may preclude the occurrence of an error of $\frac{1}{8}$ per cent, and using manufacturers' guarantees instead of handbook values gives results 0.3 per cent nearer the correct value per 5 per cent error in steam rates when computing the net benefits of extraction.

5 All of these possible errors, obtained by computation of typical cycles, illustrate the necessity of understanding turbine-extraction and stage-heating characteristics when calculating feedwater-heating cycles. To make clear the principles used in this paper and in the method which it offers, a short illustrated discussion of the simple straight-line characteristics of turbines, telling of the relation of inlet and nozzle pressures and temperatures to total steam extracted and to load carried, has been included and may be found in a later section.

Method of Computing Cycles Outlined

6 Much has been published of the benefits of feedwater heating by stage extraction, but comparatively little has been written of how those benefits are computed. Since installations in dif-

ferent power stations are rarely identical, the necessity of calculating cycles for each plant is imperative if accurate results are to be expected. A comprehensive method, utilizing no gross approximations, which may apply at one plant and not at another, but which, nevertheless, must be simple, conducive of mathematical accuracy, and adaptable to all types of cycles, is needed.

7 A method developed in the offices of the company with which the authors are associated is thought to possess these qualifications. It has not been made empirical for the sake of brevity; it attacks all details of the problem and aims to attain simplicity and accuracy only by convenient and logical arrangement of the calculations.

8 Only three fundamental processes are necessary in computing feedwater-heating economies. They are:

a Determination of the heating value of the extracted steam
b Computation of the amount to extract
c Determination of the effect of extraction upon output and, consequently, upon economy.

9 Finding the heating value of the extracted steam and its effect in decreasing output (if no additional throttle steam is supplied) involves the use of an all-important tool in this work — the turbine expansion line as drawn upon a standard Mollier chart. From this simple line can be read directly the temperature, quality, total heat, and work done at any pressure in the machine and at any fractional load with an accuracy comparable to test accuracy.

10 Once the expansion line is obtained, feed-heating calculations are a matter of simple arithmetic, but the determination of this graph from the manufacturers' guarantees is more complex. The illustrative tabular form accompanying the method later described itemizes the calculations which must be performed, and a later paragraph tells in detail, for the benefit of those not closely acquainted with turbine practice, how stage efficiencies in various types of machines are varied. Fig. 1 shows typical expansion lines of a 20,000-kw. straight-reaction unit, upon which all computations in this paper are based.

Effect of Generator and Mechanical Losses Upon Computed Heat Consumption

11 Since the turbine expansion line, as drawn upon a Mollier chart, is the basic tool with which the calculator of extraction economies works, its determination with fair accuracy is important.

12 The expansion line represents conditions as they occur inside the turbine casing; it is concerned with generator efficien-

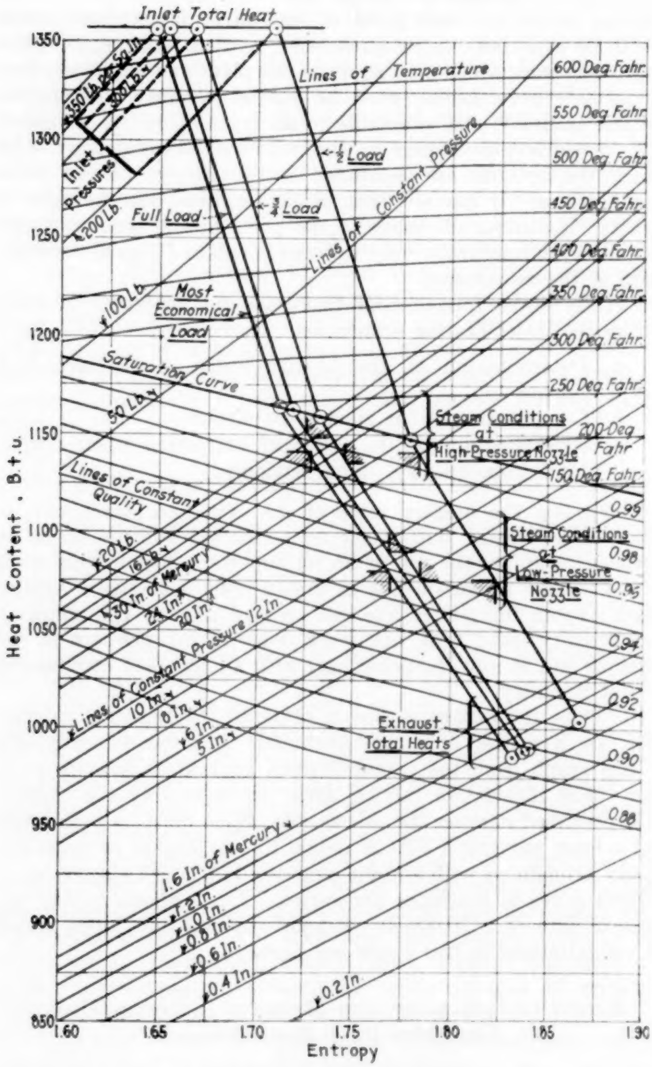

Fig. 1 Turbine Expansion Lines, Basis of Extraction Calculations

From these easily drawn lines, data for which are taken from the manufacturer's guarantees, may be read directly the quality, temperature, total heat or work done for steam extracted at any pressure, with accuracy comparable to test accuracy. All computations in this paper were based upon the above lines, representing the expansion at nominal loads of a 20,000-kw. straight-reaction turbine.

cies and mechanical losses only so far as they furnish a means of correcting manufacturers' guarantees to give the "internal" efficiencies and conditions. Obviously to charge the exhaust steam with greater heat than it actually possesses because generator or bearing losses are not taken into consideration, is a gross error.

13 Generator efficiencies form the greater portion of external losses, and if not determined from the manufacturers' guarantees must be estimated before the turbine expansion line can be drawn. Economics should determine an expression for the electrical efficiency of various-sized units, but unfortunately no simple relation can be derived which will accurately express generator losses as a functon of capacity, load, and speed. Omit-

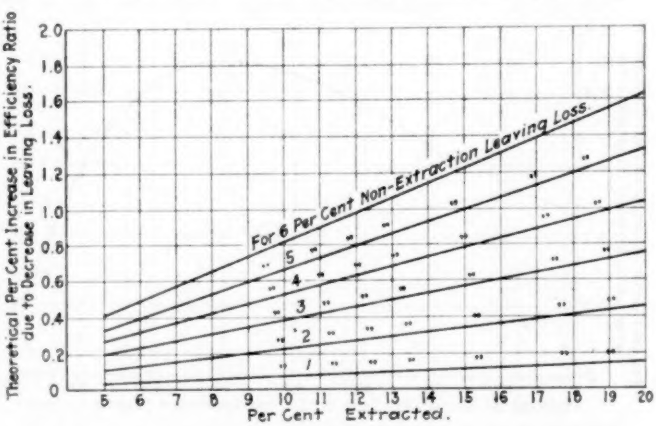

FIG. 2 DECREASE IN TURBINE LEAVING LOSS EFFECTED BY EXTRACTION

Extraction of steam helps combat the factor which has greatest influence in limiting present specific capacities. Leaving velocities through exhaust blading are reduced when the amount of steam passing them is decreased by extraction.

ting speed and other variables, the efficiencies at fractional loads of representative-sized generators used in extraction feed-heating installations are approximately represented by the empirical equation:

$$\text{Efficiency} = 0.98 - \frac{0.055}{\sqrt[3]{\dfrac{\text{Kw. Rating}}{1000}}} \times \frac{\text{Rating}}{\text{Load}}$$

Equipment supplied by different manufacturers may vary one per cent in efficiency, but as will be shown later, an error of that magnitude results in an inconsiderable error when computing extraction heat consumption.

14 Mechanical losses, which include bearing friction and gland and pump resistance, vary, as do electrical losses, as some odd function of the size of the unit. A rough rule of thumb which may be used for feedwater-heating calculations states that mechanical losses in percentage of normal rating equal

$$\frac{4.0}{\sqrt{\frac{\text{Kw. Rating}}{1000}}}$$

For a 3000-kw. machine the losses computed by this formula aggregate about 2 per cent of the normal rating; for a 30,000-kw. unit they total 0.7 per cent. In any event the absolute error in

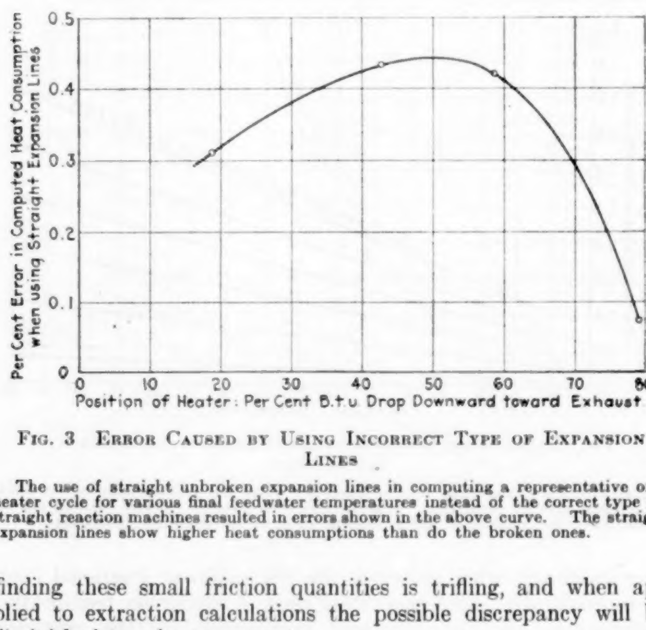

Fig. 3 Error Caused by Using Incorrect Type of Expansion Lines

The use of straight unbroken expansion lines in computing a representative one-heater cycle for various final feedwater temperatures instead of the correct type for straight reaction machines resulted in errors shown in the above curve. The straight expansion lines show higher heat consumptions than do the broken ones.

finding these small friction quantities is trifling, and when applied to extraction calculations the possible discrepancy will be diminished to minute amounts.

Use of Non-Extraction Expansion Lines to Determine Extraction Data

15 As previously stated, expansion lines are drawn from manufacturers' guarantees, though these performances represent non-extraction operation. The application is not strictly correct; however, it causes an error which is probably not discoverable by even extremely accurate computation. As shown in Fig. 5, curves 5, 6, and 7, the expansion lines of both types of operation are essentially of identical form, differing only in efficiency for

the case of machines proportioned for extraction (expansion lines 6 and 7) and in efficiency, position, and length for machines not previously so proportioned by the manufacturer (expansion lines 5 and 7). The discrepancy in final heat consumption would be caused by the very slight change in efficiency ratio due to the small change in velocity ratio, leaving loss, and vacuum, and, to continue consideration of fine details, to the lack of parallelism of vacuum and higher-pressure lines of the Mollier chart. Since studies discussed under the next topic show only $\frac{1}{16}$ per cent error in heat consumption due to 1 per cent error in the estimated efficiency ratio used when drawing expansion lines, no practical difference in total heats or temperatures taken from them at any pressure is worth consideration.

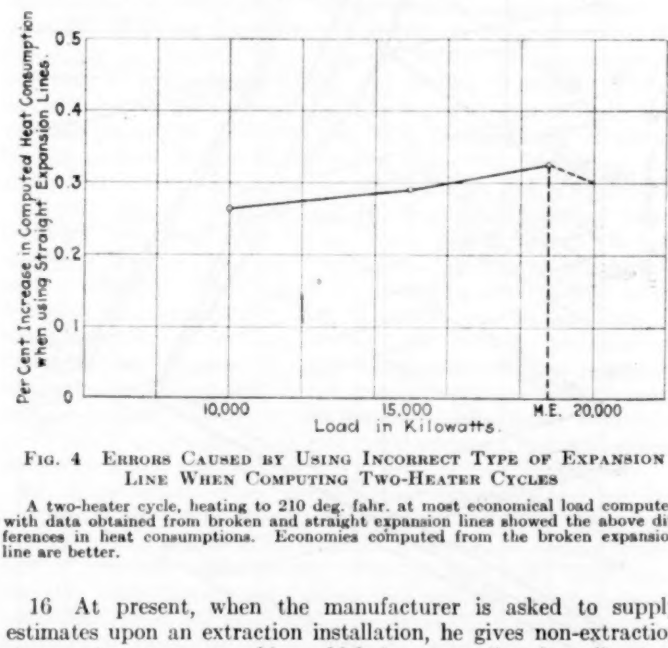

FIG. 4 ERRORS CAUSED BY USING INCORRECT TYPE OF EXPANSION LINE WHEN COMPUTING TWO-HEATER CYCLES

A two-heater cycle, heating to 210 deg. fahr. at most economical load computed with data obtained from broken and straight expansion lines showed the above differences in heat consumptions. Economies computed from the broken expansion line are better.

16 At present, when the manufacturer is asked to supply estimates upon an extraction installation, he gives non-extraction steam rates upon a machine which is necessarily of smaller turbine capacity than the machine which he would finally provide. For a given unit the most economical load position, which is determined by the amount of steam that can be passed through the machine without admitting steam at lower stages or opening up additional nozzles, differs when extracting various amounts of steam, and as a result the manufacturer must estimate upon a fictitious unit in order that his guarantees may be representative of consumptions under actual extraction operation and that the most economical load point may occur at the specified load,

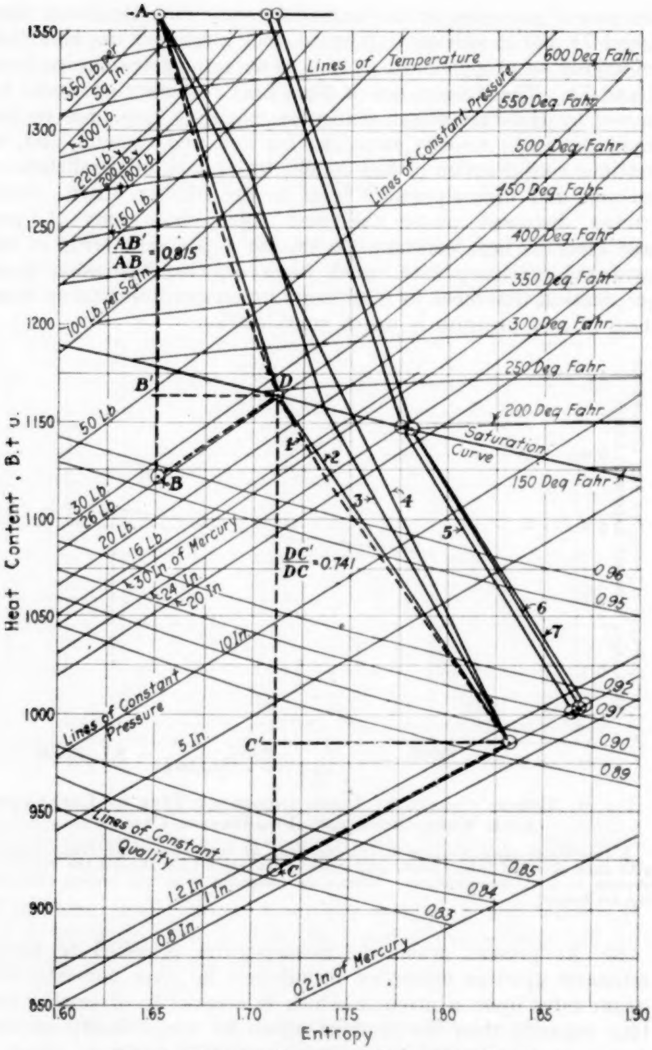

Fig. 5 Various Types of Expansion Lines

Line 1 — Test characteristics of the straight reaction-type turbine.

Line 2 — Easily drawn straight-line expansion approximating line 1, by showing 10 per cent greater efficiency in superheat field than in saturated field, gives data of high accuracy for feedwater-heating calculations.

Line 3 — Estimated expansion line of a combination velocity-compounded impulse and reaction turbine.

Line 4 — Approximate expansion in a straight impulse machine, velocity-compounded in high-pressure blading and pressure-compounded in exhaust.

Line 5 — Half-load expansion when extracting 15 per cent steam from a straight reaction turbine not previously proportioned by the manufacturer for extraction. Inlet pressure has been increased 10 lb. above the normal non-extraction value.

Line 6 — Half-load expansion when extracting steam from a unit proportioned by the manufacturer for the given percentage of extraction. Inlet pressure is identical with the value upon which the guarantees are based.

Line 7 — Expansion line when not extracting. Separation from line 6 near exhaust is due to decrease in leaving losses resulting from extraction.

Therefore expansion lines based upon such steam rates are identical in position and length for both extraction and non-extraction operations.

17 Machines not proportioned for extraction, to carry a given load when extracting, must have greater inlet pressure than the designed values, for greater weights of steam must be passed at that load. Common practice states that for every pound extracted a given fraction of a pound must be added at the throttle, but at best the rule is a gross approximation if applied indiscriminately to all nozzles. The expansion line for extraction operation of a unit not designed for extraction is represented by line 5, Fig. 5. Apparently the line differs from the non-extraction one only in position and length, and though practically it does not change the accuracy of the data which it furnishes, it does represent a change in the output for a given amount of steam. In the method of computing entire cycles later described, the method of allowing for this effect upon output will be indicated.

18 Extracting steam at large loads decreases leaving losses from the exhaust blading, for less steam passes through the last rows — with consequent lower velocities. Fig. 2 gives an approximate indication of the effect in decreasing leaving loss upon efficiency of average units, and, of course, can be used only if the non-extraction leaving losses of the unit in question are known. Good practice in large installations requires less than two per cent leaving loss at full load and 1.5 per cent at most economical load, though at 100 per cent power-factor overloads or in small units exhausting into high vacuums this percentage should be raised considerably.

19 A secondary effect of decreasing exhaust steam by extraction is the slight increase in vacuum resulting when less exhaust heat is absorbed by constant quantities of cooling water and at constant temperatures. The economy will vary considerably with different installations, and only the use of vacuum-correction curves for the particular turbine and vacuum-versus-total-steam curves for the condenser, which may be obtained from the manufacturer, will indicate the results. For an average installation the decrease in heat consumption due to increase in vacuum resulting from extraction is estimated at 0.25 per cent. Another and probably better method of estimating the gain due to having less steam pass to the condenser, is to consider the decrease in cost of providing a smaller condenser. On a cost basis the economy is 3 per cent of the initial condenser purchase price for typical equipment and extraction practice.

Influence of Stage Efficiency as Represented by the Expansion Line upon Accuracy of Feedwater-Heating Calculations

20 That the easily drawn straight-line expansion line, broken to represent approximate relative efficiencies of the high- and low-pressure stages, is fully accurate enough for the most exacting of feedwater-heating calculations, has been definitely demonstrated by a considerable number of comparative computations.

21 To magnify the results and to test for agreement over all ranges of efficiencies, theoretical and practical, broken lines with efficiencies from 90 per cent to 65 per cent were drawn and representative cycles worked with the data from each. Later, straight and broken expansion lines were compared, with results as shown by Figs. 3 and 4.

22 Comparison of straight unbroken lines with the more nearly approximate broken lines shows a maximum discrepancy of 0.45 per cent in the final computed heat consumption. Fig. 3 represents the errors at various positions of a single heater, computed for most economical load conditions, and Fig. 4 shows the maximum discrepancy in working up a more typical cycle of two heaters to be 0.325 per cent. Though inspection of the expansion lines in Fig. 5 will show a wide difference in the two types of curves, the above percentage differences resulting from their use are of a small order. They are large enough, however, to warrant the additional trouble of drawing the broken expansion lines, though assuredly not of drawing the curved expansion lines. Thus, provided the overall efficiency of the unit is correctly known, feedwater-heating data can be obtained with an accuracy commensurate with the general accuracy of engineering calculations.

23 Error in estimating the overall efficiency of the unit when drawing expansion lines results in a maximum extraction-gain error due to inaccuracy of heating calculations of $\frac{1}{16}$ of one per cent per 1 per cent error in efficiency ratio. The error decreases to $\frac{1}{40}$ per cent as the efficiency ratio decreases from 90 per cent to 65 per cent. The importance of correcting the manufacturers' guarantees for bearing friction, and of knowing generator efficiencies accurately, is therefore not great, and either values may be estimated if extreme accuracy is not desired. Both corrections, however, are quite accurately made in obtaining internal efficiencies by the method later demonstrated.

Effect of Neglecting Pressure Drop in Turbine Due to Extraction and to Drop in Piping to Heater

24 Pressure drops decrease saturation temperatures and therefore, when occurring at the high-pressure heater of a given system, are particularly influential upon economy since they de-

crease the final temperature to which the feedwater may be heated.

25 Ignoring the drop at the turbine nozzle due to extraction when computing a two-heater cycle heating to 210 deg. fahr. at the most economical load, causes errors of 0.06 per cent, 0.07 per cent, and 0.1 per cent at ½, ¾, and most economical loads, respectively. These values correspond roughly to a 5 per cent drop in pressure, and can be applied in case of drops through pipe lines to heaters. In both cases the computed heat consumption is too small if the drops are not considered.

Error in Neglecting Flashing of Drained Condensate

26 Flashing of drained condensate from one heater into the hotwell of the next lower one of course involves a thermodynamic loss, but reliability, investment, and operating considerations do

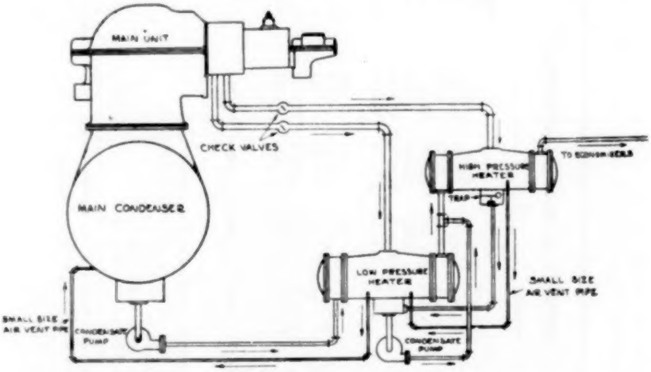

Fig. 6 Flow Diagram of a Simple Two-Heater System

Computation of feedwater cycles is clarified and expedited by use of sketches allowing the computer to visualize all processes in the cycle.

not favor the installation of additional pumps to discharge the drained condensate from each heater into the feedwater line above that heater. Computations show flash loss as follows: For a two-heater cycle heating to 210 deg., 0.03 per cent; three heaters to 240 deg. fahr. and 280 deg. fahr., 0.07 per cent and 0.20 percent, respectively; and four heaters to 280 deg., 0.095 per cent. Allotting the greatest share of the heating load to the low-pressure heater and increasing the number of heaters tend to minimize flash losses.

27 In determining the heat available in the extracted steam to heat feedwater its total heat minus the heat of the liquid at the particular heater pressure is the correct value. More of its heat is later available in the hotwells of succeeding heaters as

COMPUTATION OF FEEDWATER HEATING CYCLE

Steam Conditions:- 350 lb. gage; 250 deg. fahr.; 29 in. vacuum
Equipment in Cycle:- Two heaters

No.	Item	Fractional Load 1/2	Fractional Load 3/4	Fractional Load M.E.	Full	High-Pressure Heater, Approx. Fractional Loads 1/2	High-Pressure Heater 3/4	High-Pressure Heater M.E.	Low-Pressure Heater, Approx. Fractional Loads 1/2	Low-Pressure Heater 3/4	Low-Pressure Heater M.E.
	Non-Extracting Operation										
1	Output, kw. .... (given)	10,000	15,000	18,680	20,000	..	..	..	..	..	..
2	Steam Rate, lb. per kw-hr. ...... (given)	10.18	9.65	9.53	9.57	..	..	..	..	..	..
3	Total Steam, lb. per hr. (No. 1 × No. 2)	101,800	144,800	178,000	191,400	..	..	..	..	..	..
4	Generator Efficiency .... (given)	0.948	0.960	0.964	0.966	..	..	..	..	..	..
5	Brake Horsepower Output [No. 1 ÷ (No. 4 × 0.746)] .	14,120	20,920	26,000	27,750	..	..	..	..	..	..
6	Total Heat at Throttle, B.t.u.per lb.(From steam table)	1356.5	1356.5	1356.5	1356.5	..	..	..	..	..	..
7	Work Done, B.t.u. per lb. [(No. 5 × 2545 ÷ No. 3) + °/o for friction]	353	368	372	369	..	..	..	..	..	..
8	Total Heat at Exhaust, B.t.u. per lb. (No.6 minus No.7)	1003.5	988.5	984.5	987.5	..	..	..	..	..	..
9	Heat of Liquid at Hotwell, B.t.u. per lb. (Steam table)	47	47	47	47	..	..	..	..	..	..
10	Heat Supplied by Boiler, B.t.u.per lb.(No.6 minus No.9)	1309.5	1309.5	1309.5	1309.5	..	..	..	..	..	..
11	Heat Consumption, B.t.u. per kw-hr. (No. 2 × No. 10) ..	13,331	12,637	12,482	12,533	..	..	..	..	..	..
12	Inlet Pressure, lb.abs. (M.E. and full-load values = throttle pr. abs. - 5°/o and 7°/o resp.; 1/2 and 3/4 load values are proportional to No. 3,M.E.load)	200	284	350	340	..	..	..	..	..	..
	Extracting Operation										
13	Pressure at Nozzle, lb. abs. (M.E.load value given; others proportional to No. 3).........					10.05	14.3	17.4	2.81	4.0	4.9
14	Pressure at Heater, lb. abs. (No. 13 minus 5°/o pipe drop and 2°/o extraction drop at h.-p. nozzles; 5°/o extraction drop at l.-p. nozzles)					9.35	13.3	16.2	2.53	3.6	4.4
15	Saturation temperature at Heater Pressure, deg. fahr. (From steam table)					190	207	217	135.	149	157
16	Final Feedwater Temperature [No. 15 minus terminal difference (7 deg.)]					183	200	210	128	142	150
17	Heat of Liquid of No. 16, ...					151	168	178	96	110	118
18	Heat Supplied by Boiler, B.t.u. per lb. [No. 6 minus No. 17 (for h.-p. heater only)].....					1205.5	1188.5	1178.5	..	..	..
19	Total Heat at Nozzle, B.t.u. per lb. (From Mollier expansion lines)					1137	1137	1141	1076	1076	1079

20	Total Heat at Exhaust, B.t.u. per lb. (No. 8)	1003.5	988.5	984.5	1003.5	988.5	984.5
21	B.t.u. per lb. Work Lost in Extracted Steam (No.19 minus No. 20)	133.5	148.5	156.5	72.5	87.5	94.5
22	Kw-hr. Work Lost per 1000 lb. per hr. of Steam Extracted $\left(\frac{\text{No. 21} \times \text{No. 4} \times 1000}{3416}\right)$	37.0	41.7	44.3	20.1	24.6	26.7
23	Total Heat of Steam at Each Heater, B.t.u. per lb. (No. 19 minus radiation losses)	1137	1137	1141	1079	1076	1076
24	Heat of Liquid of Steam at Each Heater, B.t.u. per lb. [No. 17 + term. diff. (7°); for l.-p. htr., No. 9]	158	175	185	47	47	47
25	Heat Available for Heating Feedwater, B.t.u. per lb. (No. 23 — No. 24)	979	962	956	1032	1029	1029
26	Final Temperature of Feedwater, deg. fahr. (No. 16)	183	200	210	128	142	150
27	Initial Temperature of Feedwater, deg. fahr. (No. 26 of lower-pressure heater)	128	142	150	79	79	79
28	Temperature Rise of Feedwater, deg. fahr. (No. 26 minus No. 27)	55	58	60	49	63	71
29	Quantity of Feedwater, lb. per hr. [No. 3 (except for l.-p. heater = No. 3 minus sum of No. 33 for all higher-pressure heaters)]	101,800	144,800	178,000	96,120	136,140	166,910
30	Heat Transferred to Feedwater, million B.t.u. per hr. (No. 28 × No. 29 ÷ 10^6)	5.6	8.4	10.68	4.72	8.57	11.84
31	Heat Flashed into Heater by Drained Condensate, million B.t.u. per hr. (No. 28 × sum of No. 33 of all higher-pressure heaters)	..	..	..	0.31	0.50	0.66
32	Net Heat Supplied by Extracted Steam, million B.t.u. per hr. (No. 30 minus No. 31)	5.6	8.4	10.68	4.41	8.07	11.18
33	Weight of Steam Extracted, lb. per hr. [No. 32 × 10^6) ÷ No. 25]	5,720	8,720	11,170	4,280	7,850	10,870
34	Kw. Work Lost in Extracted Steam, kw-hr. (No. 33 ÷ 1000 × No. 22)	211	364	495	86	193	290
35	Total Kw. Decrease in Output, kw-hr. (Sum of No. 34 for all heaters)	297	557	785	..	..	..
36	Kw. Output Extracting (No. 1 minus No. 35)	9,703	14,443	17,894	..	..	..
37	Per Cent Decrease in Output when Extracting (No. 35 ÷ No. 1)	2.96	3.7	4.18	..	..	..
38	Steam Rate, lb. per kw-hr. (For exact fractional loads of units proportioned for extracting, No. 3 ÷ No. 36)	10.490	10.025	9.948	..	..	..
39	Steam Rate, lb. per kw-hr. (For non-proportioned units, from curve of No. 38 vs. No. 36 at exact fractional loads)	10.437	9.996	9.948	..	..	..
40	Heat Consumption, B.t.u. per kw-hr. (No.18 × No. 38 if extraction unit, No. 18 × No. 39 if not proportioned)	12,583	11,880	11,724	..	..	..
41	Weight Extracted, Fractional Loads, Each Heater, lb. per hr. (No. 33 + °/o given in No.37)	5,850	8,980	11,550	4,420	8,150	11,350
42	Weight Extracted, Fractional Loads, All Heaters, lb. per hr. (Sum of No. 41)	10,270	17,130	22,900	..	..	..
43	Total Steam, lb. per hr. (No. 3 plus °/o given in No. 37)	104,700	150,000	185,000	..	..	..

FIG. 7 SAMPLE CALCULATION OF A TWO-HEATER CYCLE

In making complete calculations, steam-jet air pumps, deaërators, evaporators, and steam-driven auxiliaries are given columns just as are the heaters.

(Sample forms for computing the feedwater heating cycle as above may be procured upon application to the Secretary of the Society, 29 W. 39th St., New York)

may be realized by inspection of Fig. 6, but computation of this amount should be made in flash calculations.

Details of Authors' Method of Computing Cycles

28 Based upon the initial assumption that non-extraction total steam is passing through the turbine, the method considers the effect of extraction upon output and then later adjusts the results to full extraction operation at the even fractional loads. It may be considered as an estimate of temperatures, pressures, and quantities which would occur at every point of the cycle were tests made upon the turbine and its auxiliaries, first at normal loads with its extraction nozzles closed, then at decreased loads

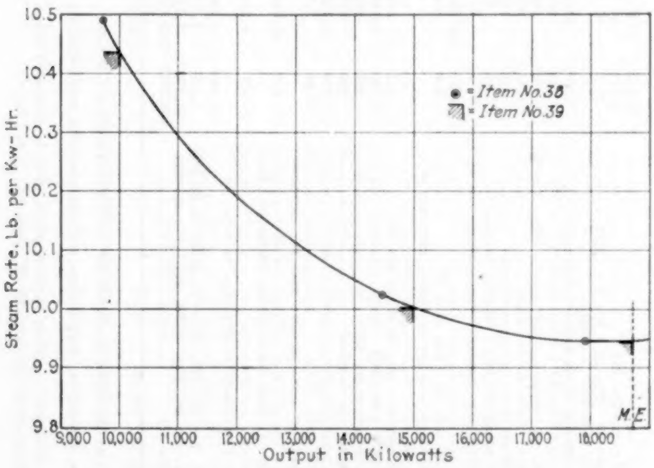

Fig. 8 Reduction of Extraction Steam Rates at Decreased Loads to Rates at Exact Nominal Loads

The method developed through considerable use in the offices of the company with which the authors are associated first considers extraction operation at loads reduced because of extraction and later solves, with the use of the above curve, for full nominal-load values.

after the nozzles are opened, and finally at full loads again while still operating extracting.

29 As shown in the sample computed cycle, Fig. 7, the calculations are tabular, the first part being devoted to straight non-extracting operation and the latter to extraction. The function of the entries in the columns will now be explained.

30 Non-extraction operation is useful in indicating the heating value of the extracted steam and the effect of its extraction upon output. Items 1 to 8 and item 12 give data for drawing expansion lines, for they determine exhaust and inlet points, which,

when properly plotted and connected upon a Mollier chart, result in a curve telling the pressure, temperature, quality or superheat, and total heat of the steam at any point in the machine. Exhaust total heat is plotted on the 1-in. absolute-pressure line, the vacuum in this problem being 29 in., and inlet pressure is plotted upon the 1356.5-B.t.u. total-heat line.

31 Tests, supported by thermodynamic analysis, have shown straight reaction-type turbines to have 10 per cent greater efficiency in the superheat field than in the saturated area. To represent these test conditions a broken line connecting the inlet and exhaust points, and changing direction at the saturation line,

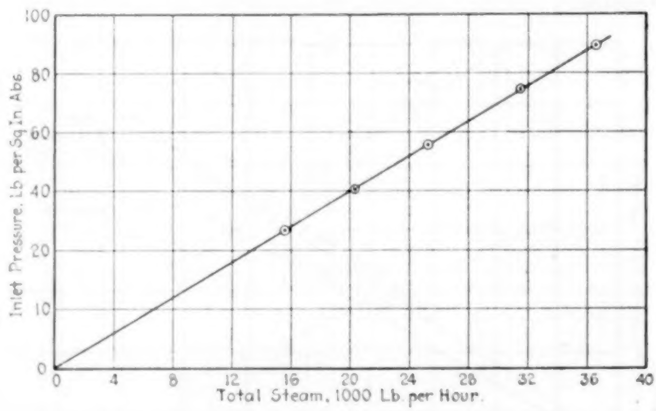

Fig. 9 Inlet-Pressure Characteristics of a Turbine

Even reaction-type machines behave much as large orifices. Total steam passed is always proportional to inlet pressure.

serves very well. Actually the line should be curved, as shown in the figure, but such refinements for extraction calculations are unnecessary.

32 To draw lines representing the stated difference in efficiency ratio requires cut-and-try approximations until the correct values are reached. For the initial trial, step off to the left of a straight line connecting inlet and exhaust points about an inch (on the full-sized Mollier chart)[1] on the saturation curve, and through that point draw a temporary broken expansion line. The stage efficiency represented by each part of the line equals the work actually done in each stage (*AB′* and *DC′*, respectively, in Fig. 5) divided by the work possible in a perfect turbine, represented by *AB* and *DC*, respectively. When the ratio of these two efficiencies equals 1.10, as in the figure shown, then the lines may be permanently drawn and used for obtaining extraction data.

[1] About $\frac{5}{16}$ in. on the reduced chart, Fig. 5.

33 Combination impulse and reaction machines have expansions appearing somewhat as is shown by curve 3, Fig. 5, due to the inherent inefficiency of the impulse element used. Unless the heat drop and efficiency of that element are known, an expansion line drawn as previously described but on the basis of 5 per cent efficiency difference in the superheat and saturated fields, is advised. Straight impulse-type machines, velocity-compounded in the high-pressure stages and pressure-compounded at the exhaust end, have more nearly a straight-line expansion characteristic and may be drawn as such, though of course manufacturers' test results should always be used in drawing expansion lines if available.

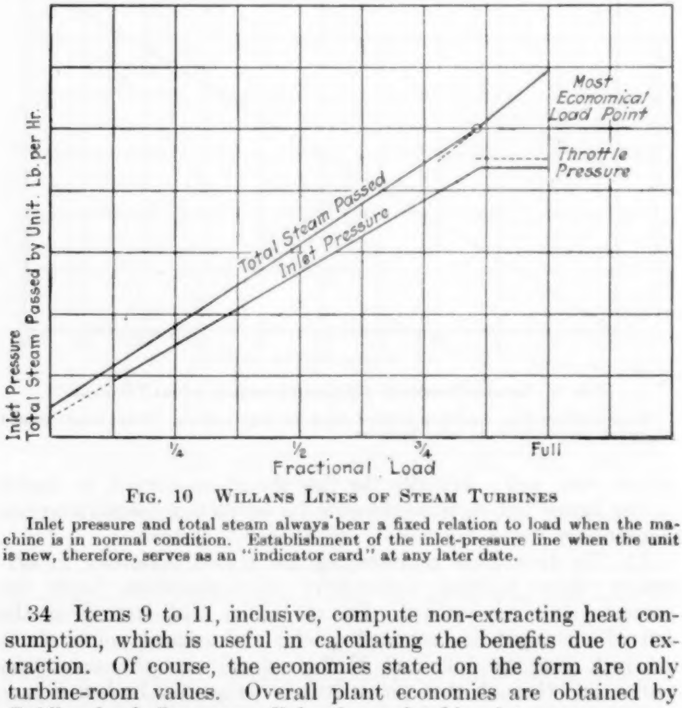

Fig. 10 Willans Lines of Steam Turbines

Inlet pressure and total steam always bear a fixed relation to load when the machine is in normal condition. Establishment of the inlet-pressure line when the unit is new, therefore, serves as an "indicator card" at any later date.

34 Items 9 to 11, inclusive, compute non-extracting heat consumption, which is useful in calculating the benefits due to extraction. Of course, the economies stated on the form are only turbine-room values. Overall plant economies are obtained by dividing by boiler-room efficiencies and taking into account consumption of auxiliaries and wastages.

35 Under the caption "Extracting Operation" the items may be divided into three sections. Items 13 to 25 give preliminary values which are later used in the actual cycle calculations. Items 26 to 36 solve for the decrease in output resulting from extraction. The remaining items give economies and quantities when full fractional loads are being carried. The notes after each

entry, inserted for the purpose of expediting computation and diminishing chances of gross errors, show how the values are obtained.

36 All values, with one exception, have physical meanings, and when one calculates a cycle using the sample tabular form he should be able to follow calculations with a mental picture justifying all mathematical operations. The one exception is in item 24, low-pressure heater, in which a subterfuge is undertaken to eliminate the necessity of a cut-and-try calculation in obtaining the last-heater value of item 29. Net results are exactly the same, but the computer should not attempt to find a physical basis for routing part of the drained condensate to the condenser and the remainder normally as shown in the flow diagram of Fig. 6.

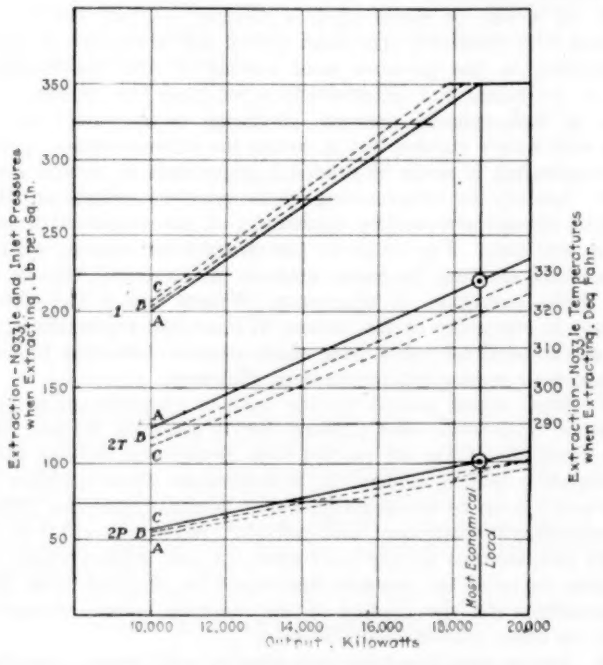

Fig. 11 Extraction Characteristics

Pressures and temperatures fall according to straight-line functions with decreasing loads and amounts extracted.

Curves 1-*A*, *B*, and *C* represent inlet pressure when operating non-extracting and when extracting 5 per cent and 10 per cent, respectively.

Curves 2*T*-*A*, *B*, and *C* show extraction-nozzle temperatures when operating non-extracting and when extracting 5 per cent and 10 per cent, respectively.

Curves 2*P*-*A*, *B*, and *C* indicate pressures at a 100-lb. [non-extraction pressure most economical (M.E.) load] nozzle for the conditions cited above.

37 If turbines are thought of as simple orifices, their straight-line extraction characteristics as utilized in the form will be more easily understood. Though turbines consist of a multiplicity of small orifices of varying sizes and mounting, part of which in reaction-type machines are moving, they behave much the same as single large nozzles. Turbine terminology speaks of machines "passing" given amounts of steam at certain inlet pressures, and in fact, experience has shown that simple nozzle formulas apply to machines commonly thought of as non-nozzle machines, with practical accuracy.

38 If a test be made upon a straight reaction machine at various inlet pressures, the steam passed will always be in direct proportion to the pressure head forcing it into the machine. Fig. 9, an actual test determination, indicates this relation. In case of high exhaust pressure, discharge is decreased just as it is with simple nozzles, but of course the influence when operating condensing is never of practical importance at normal loads.

39 Analogy to reciprocating-engine practice affords an illustration, though not strictly applicable, of the variation of total steam and load. Fig. 10 shows the straight-line relation existing in turbines, showing the same intercept characteristic though not the slight curvature of the engine Willans line. The abrupt change in curvature of the turbine Willans line represents opening of the overload valve, past which steam is admitted to lower stages, with consequent decrease in efficiency.

40 Since steam passed by the unit is proportional to inlet pressure, load-versus-inlet-pressure curves resemble Willans lines. The lower line of Fig. 10, plotted from representative tests, shows this relation, which, incidentally, if determined when machines are new and known to be operating under normal conditions, affords an exceedingly convenient and reliable "indicator card" of the unit's performance at any later time. A line slightly above the original curve of the pressure line would be obtained from later observations did the blading of the machine become eroded or dirty or other unseen troubles occur.

41 Just as inlet pressures vary directly with steam passed and as given straight-line functions of the load carried, so do the pressures at any point in the machine vary. The variation of pressure at a 100-lb.-absolute extraction point provided on a standard machine is shown by curve *2P-A*, Fig. 11, which indicates the pressure decreasing in straight-line relation with the load. Temperatures vary in the same manner, as shown in curve *2T-A*. Therefore the temperature to which feedwater can be heated by steam extracted from any given nozzle is a function of the load carried.

42 Another influence affecting nozzle temperature and pressure is the amount of steam being extracted. If the blading past the nozzle be thought of as a separate turbine and the extraction-

nozzle pressure as its inlet pressure, then the reason for a decrease in pressure when extracting will be appreciated. The decrease for given loads will not be directly proportional to the amount extracted, for to carry the same capacity when extracting, inlet pressures must be increased. The relation of extraction- to non-extraction-nozzle pressure is therefore dependent upon the position of the nozzles under consideration as well as the amount extracted, since the necessary increase in inlet pressure is proportional to the amount of work lost when steam is extracted. Considerable work is lost when extraction is done at high pressures; little at low pressures. Fig. 11 shows the relation of inlet pressure and extraction-nozzle pressure and temperature when extracting various amounts.

43 Summarizing, turbine principles state that:

a Total steam passed is directly proportional to inlet pressure

b Inlet pressure varies according to a straight-line-with-intercept relation to load

c Extraction-nozzle pressures bear the same relation to load as does inlet pressure

d Extraction-nozzle temperatures, varying with nozzle pressures, also bear the same relation to load

e Nozzle pressures and temperatures decrease proportionally with the amount extracted, for given loads by amounts dependent upon the position of the nozzle.

44 These obvious principles, containing nothing which could not be found in turbine textbooks, have been noted in this discussion because of their important bearing upon the subject of extraction. As stated previously, knowledge of these fundamentals and ability to use the Mollier chart should allow any power-plant man to compute an extraction feedwater-heating cycle for his plant.

Application of Method

45 In actual use, separate blueprints, photostats, or other reproductions of the tabular form Fig. 7 are pasted to the calculation pages and separate solutions made for each cycle. Ordinary five-line-per-inch paper, properly ruled vertically for the specific cycle, is best for the calculations. With some experience, the calculations may be performed with great rapidity, many values being obtained without changing the slide-rule setting. Given only guarantees, generator efficiencies, and extraction-nozzle pressures, a representative two-heater cycle may be worked in two or three hours by any one slightly acquainted with the work.

46 Slide-rule accuracy is fully accurate enough for the most exacting requirements in all except two items, items 38 and 40.

In the preliminary computations for these values, small quantities are used throughout, giving slide-rule readings that are accurate to the last kilowatt. In making the division for item 38 and the multiplication for item 40, solution by comptometer, longhand, or a combination of longhand and slide rule is necessary if accuracy greater than 0.1 per cent is desired. The latter method of combining longhand and slide-rule calculations has been found particularly adaptable.

47 Solution of cycles containing steam-jet air pumps, deaërators, evaporators, steam pumps or other auxiliary equipment does not lessen the value of the tabular form, for this equipment is given columns in the calculations just as are the heaters. The solution is in the opposite direction, however, since total steam

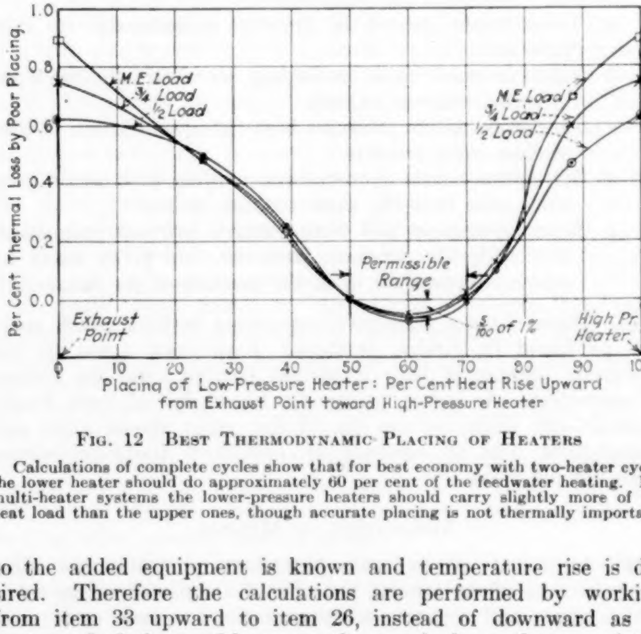

Fig. 12 Best Thermodynamic Placing of Heaters

Calculations of complete cycles show that for best economy with two-heater cycles the lower heater should do approximately 60 per cent of the feedwater heating. For multi-heater systems the lower-pressure heaters should carry slightly more of the heat load than the upper ones, though accurate placing is not thermally important.

to the added equipment is known and temperature rise is desired. Therefore the calculations are performed by working from item 33 upward to item 26, instead of downward as in heater calculations. Of course, the terminology of some of the items must be mentally broadened to apply to all types of equipment. Cycles containing desuperheaters are computed in a manner no different than are systems without that equipment, though of course item 43 must be increased by the amount of injection water if such is supplied from an external source. If evaporation of make-up water is intermittent, occurring only at periods of low loads and storing for full loads, as seems advisable practice in future designs, then the calculation of evaporators should be applied only at fractional loads.

48 Allowance for power to drive electrical auxiliaries, if that power is taken from the main unit, may be made in item 35 so that exact nominal loads at either switchboard or main bus will be obtained in the correction curve of item 39. Since full rated load (20,000 kw. in the example cited) is not obtainable on the main bus when furnishing auxiliary load without carrying a generator overload, the full-load entry must be made with power-factor qualifications.

49 For computing comparative cycles, as is done in power-plant design, the method is flexible in that it does not involve the derivation of separate formulas for each cycle as is necessary if accuracy is desired when using the formula methods. Provided a formula is used a great many times and the computer is acquainted with its notation, then it possibly is a labor-saving device.

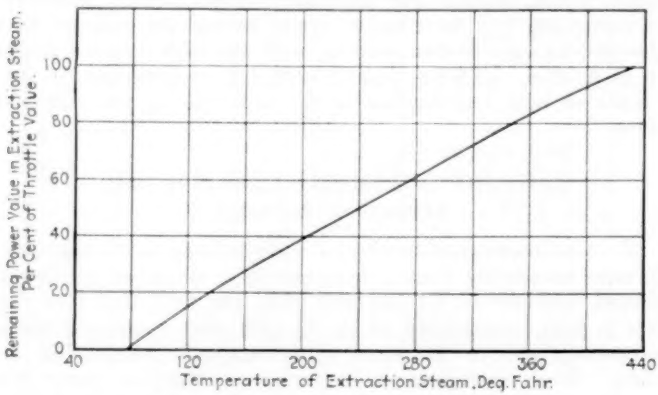

Fig. 13 Commercial Value of Steam at Various Parts of a Turbine

Curve showing why extraction at low temperatures affords inexpensive steam; and why the plan of heating feedwater by bearing oil and generator cooling air has a competing method because of the cheapness of heating by extraction.

II—MAXIMUM REALIZATION OF EXTRACTION GAINS

Effect of Relative Position of Heater upon Economy of Feedwater Heating

50 For this study a representative cycle of two heaters, heating to 210 deg. fahr. at most economical load, was initially chosen. The low-pressure heater was varied in position from 25 per cent to 87 per cent of the heat-drop interval between the high-pressure heater and the exhaust, and the resulting economies figured.

51 Fig. 12 shows that best economy is obtained when operating with the low-pressure heater at a point 60 per cent of the heat-drop interval upward from the exhaust, or in this case at a pressure of about 4 lb. per sq. in. absolute. Under these conditions slightly more heat is transferred in the low-pressure heater, though considerably more power is lost in steam extracted from the high-pressure nozzle. Accurate placing of heaters is not of importance, for, as the curve shows, any position from 50 to 70 per cent of the heat drop causes a change in economy no greater than $\frac{1}{20}$ of 1 per cent. In three- or four-heater cycles slight preference should be given low-pressure heaters in distributing the heat loads, working the high-pressure ones at lighter values, though this adjustment is not strictly necessary. In computing two three-heater cycles having the ratio of heat absorbed in each heater, starting with the high-pressure heater in each case, 1-1.6-1.1 and 1.0-0.97-1.8, respectively, a difference in heat consumption of the order of $\frac{1}{100}$ per cent was found.

Influence of Auxiliary Equipment upon Extraction Economy

52 A representative two-heater cycle, heating to 210 deg. fahr. at most economical load and containing a steam-jet air pump, showed increases of 1.1 per cent, 0.84 per cent, and 0.71 per cent in heat consumption at ½, ¾, and most economical loads, respectively, above the same cycle computed without an air pump. True, average steam consumption of the air pump (0.8 per cent of full-load throttle total steam) heated the feedwater as much as 15 deg. fahr., but this being done at an extremely low temperature level by high-temperature-level steam resulted in decreasing the benefits of extraction heating from, for example, say, 5.5 per cent to 4.5 per cent. The fallacy of the time-worn expression crediting steam-jet air pump, injector, and other steam auxiliary equipment in power stations with 100 per cent efficiency is thus strikingly indicated.

53 Were two-stage steam-jet air pump equipment so constructed that an extraction heater could be interposed between the two effects, these losses could be lessened to 0.96 per cent, 0.725 per cent, and 0.64 per cent since most of the air-pump aftercooler heating could be done at a higher temperature level. If mechanically practical and fully as reliable as the steam-jet air pump, economical motor-driven vacuum pumps, with air pumps in reserve as "augmenters" or as auxiliaries when starting, would be desirable equipment, though continuity of service is of great importance in this consideration.

Effect of Type of Turbine upon Extraction Gains

54 Since extraction gains are caused by obtaining work from the extracted steam before it passes to the heater, the amount of work done in the machine down to a given pressure is a criterion of extraction efficiency. Machines with inherently inefficient high-pressure stages are at a disadvantage, therefore, in producing economy in extraction feed-heating systems. Expansion lines 3 and 4, Fig. 5, when used in giving data for a two-heater cycle heating to 210 deg. fahr. at the most economical load, give results 0.24 per cent and 0.325 per cent, respectively, poorer in heat consumption than curve 2. The latter has high-

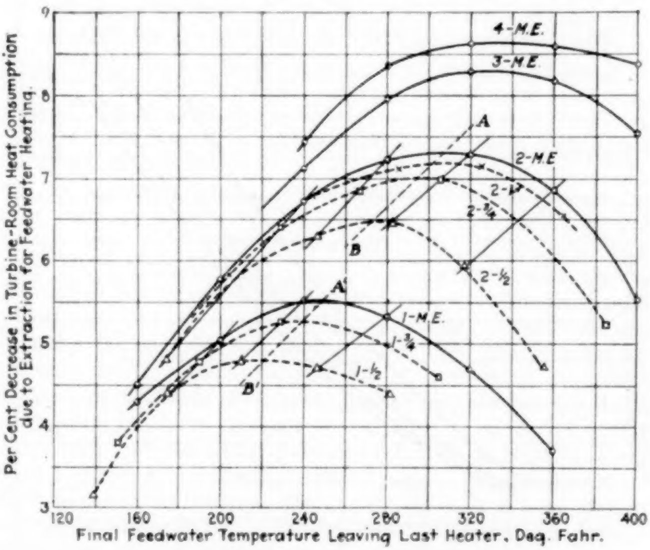

Fig. 14 Percentage of Heat Saving Effected by Extraction Heating to Various Temperatures with Different Numbers of Heaters, as Compared with Non-Feed-Heating Operation

Results of accurate computations of complete cycles, considering all extraction variables (with exception of leaving loss) in a typical installation, showing net and comparative turbine-room gains for a representative large-unit high-pressure machine, though applicable to others with small error. Curve 1-M.E. is for a one-heater cycle, most economical load; Curve 2-¾ is for a two-heater cycle at three-quarter load, etc. Data from Fig. 1.

efficiency high-pressure blading and moderate-efficiency low-pressure blading; the former, to indicate comparable overall efficiencies, must have very high-efficiency low-pressure blading to counteract inevitable losses of the high-pressure blading. The extracted steam experiences these losses, but leaves the machine too early to compensate for them in the exhaust blading, with the net results as indicated above.

Commercial Value of Heat in Generator Air and Bearing Oil

55 Due to the low temperature level of the heated generator air and bearing oil, their utilization for purely thermal considerations is questionable so long as feedwater is heated by stage extraction. As shown by Fig. 13, extraction steam taken from a 120-deg. nozzle (the average temperature of generator air and bearing oil) has given up 85 per cent of its power value in doing work before being extracted, and any heating it may do is at a cost of only 15 per cent of its value at the throttle.

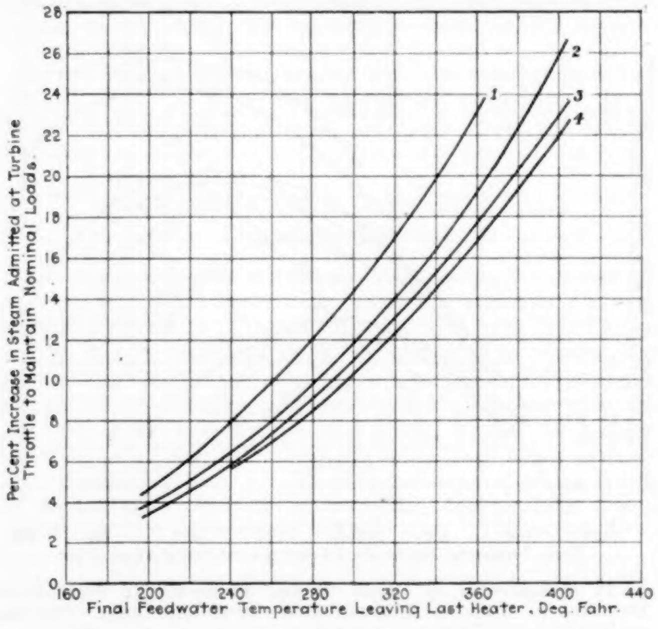

Fig. 15 Additional Throttle Steam Required to Maintain Nominal Loads When Heating to Various Temperatures by Different Numbers of Heaters

For use with Fig. 14 in determining most commercially economical heater installation in a given plant; applies for most economical loads, though only approximate at high temperatures, due to change in steam rate past normal most-economical-load point.

Four per cent mechanical and electrical losses, originally produced with a thermal efficiency of not more than 25 per cent, represent 1 per cent of the total heat, or enough to heat the feedwater about 12 deg. fahr. The same temperature rise could be obtained by extracting 1.2 per cent of throttle total steam from the 120-deg. nozzle, causing a maximum heat-consumption

increase of in the neighborhood of 0.2 per cent. Whether the more costly apparatus required for heat transmission from oil and air warrants substitution for the simpler heating by extraction steam in heaters already provided is questionable, unless of course the additional apparatus serves the dual purpose of cooling and heating as in closed generator cooling-air systems.

Comparative Economy of Stage Heating to Various Temperatures and with Different Numbers of Heaters

56 Allowing for terminal difference, extraction and pipe pressure drops, flashing, and using all refinements in cycle computations, representative systems heating to different maximum temperatures with one, two, three, and four heaters were accurately computed. In Fig. 14 the results are compared with heat consumption for non-extraction operation, though the curves afford intercomparison as well.

57 In agreement with previous published investigations, the curves show maximum benefits to occur at 245, 300, 320, and 330 deg. fahr. when heating with one-, two-, three-, and four-heater cycles, respectively. They also represent comparative economies in percentage of heat consumption at any final temperature and for any practical number of heaters. Fractional-load curves for the one- and two-heater cycles show the comparative economies at full and reduced loads, indicating maximum economy for all loads obtainable, not with constant feedwater temperature, but with changes just as they occur at extraction nozzles with variable loads, represented by the straight lines adjoining the calculated points in both systems. It will be appreciated that besides cutting the peaks of each set of curves, the straight lines AB and $A'B'$ also indicate the natural drop in temperature at the specific nozzle heating feedwater to those most efficient temperatures. Unless boiler-room practice rigidly requires constant feedwater temperatures at economizer inlets, the inherent simplicity of extraction stage heating can be maintained in the future.

58 Increase in the number of heaters, besides improving economy at any given final temperature and increasing the most economical temperature, broadens the range over which the system can operate without causing great decrease in economy. Three- and four-heater-cycle curves are appreciably flatter than those of fewer heater cycles. It will also be noted that the proportionate gain in adding additional heaters decreases rapidly, especially at the lower temperatures in use at present. Increase in final turbine-room feedwater temperatures, which is not unlikely, will increase the commercial advisability of providing greater numbers of heaters.

The Future of Stage Extraction for Feedwater Heating, and Conclusions

59 The use of combustion-air economizers, adoption of the unit system, improvement in the reliability of auxiliaries, and the growing demand for simplicity in power-plant construction all point toward the steady adoption of the stage-extraction method of heating feedwater.

60 That air preheaters can effectively lessen the economical rating of feedwater economizers, thus allowing greater benefits of extraction feedwater heating to accrue by heating to higher temperatures in the turbine room, seems probable. Mechanically and thermally successful air heaters have been tried and tested with creditable results, decreasing chimney temperatures to exceedingly low amounts while carrying part of the economizer heat loads. Increasing the turbine-room feedwater temperature to 300 deg. fahr. from the present practice of 210 deg. fahr. is therefore not improbable. The change would decrease economizer corrosion, increase boiler rating, and allow a gain in overall plant economy of a magnitude determinable by use of Fig. 14. The increase in feedwater temperature mentioned above would additionally improve turbine-room heat consumption 1.25 per cent if a two-heater system were in use; and 1.95 per cent for a three-heater system.

61 In smaller installations the plan of motorizing the auxiliaries, coupled with the inherent self-regulation of feedwater temperature in extraction systems, may render steam-turbine rooms more or less automatic in control. Auxiliaries can be permanently tied electrically to the main or auxiliary generator, gaining in speed with it when starting and being automatically shut down upon stopping. Feedwater heating by other means, because of the attention necessary for approximate constant-temperature maintenance and of the attention to auxiliary turbines, does not adapt itself to automatic control. Where economy considerations do not demand extraction heating, ease of operation may be a factor contributing to its installation.

62 Larger installations with positive sources of current for auxiliary operation need maintain no inefficient steam-driven auxiliaries, with perhaps the exception of a few stand-by boiler-feed pumps, and thus may realize the full benefits of stage extraction. With the increasing interlinkage of power stations, steadier line conditions are assured, promising greater reliability of motorized auxiliaries in cases where auxiliary generators are not employed. By its nature, feedwater heating by stage extraction appears to adapt itself well to present power-plant practice, and continued adoption is anticipated.

63 Accuracy in extraction calculations can hardly be expected with the use of simple equations representing the involved

internal characteristics of modern turbines. These characteristics must be understood, and for their expression the Mollier expansion line seems best suited. In calculating resulting heat consumption, allowance for all conditions in the cycle is demanded if accuracy commensurate with present requirements is desired. It is hoped that this paper may at least provide suggestions of some use to the producers of power.

No. 1913*d*

REHEATING IN CENTRAL STATIONS[1]

By W. J. Wohlenberg,[2] New Haven, Conn.

Associate-Member of the Society

In this paper the author discusses the influence of amounts of energy added in reheating, number of reheating stages, and points in expansion at which reheating should begin. A comparison is made of reheating, regenerative, and combination cycles combining both reheating and bleeding stages, and it is shown that, because of the influence on the internal machine efficiency thereby, reheating properly applied may lead to higher efficiencies than bleeding. It is furthermore shown that the combination cycles give promise of realization in practice of appreciably higher efficiencies than would be the case for the other cycles investigated. The analysis also indicates that for a limiting steam temperature of 750 deg. fahr., the maximum efficiency is attained at initial pressures varying between 600 and 900 lb., the exact optimum depending upon cycle, fuel, and steam generator used; and that if pressures are increased beyond these points there will result an actual reduction in overall efficiency.

IN AMERICA some work of this nature has been carried on by the Detroit Edison Company, but little concerning it has been published. The subject was suggested by Mr. George A. Orrok, Mem. A. S. M. E., and what is generally known of the present state of the art is well expressed in a letter from Mr. Orrok containing the following paragraph:

> There are a few stations where reheating has been done, but there is no information regarding them. The particular station where the most work has been done is the North Tees station outside of Newcastle, England, but there have been no facts of performance made public about this plant. The Chicago Edison is putting in such an outfit which they hope to have running next year. So you see you have a virgin field to work in and no precedent except your thermodynamic principles.

This paper is accordingly first occupied with an attempt to discover the proper place of reheating in the central station on an ideal basis. Secondly, modifications of ideal conditions leading to expected practical performance are introduced.

[1] For discussion and closure see pp. 766 and 814.

[2] Assistant Professor Mechanical Engineering, Sheffield Scientific School, Yale University.

Contributed by the Power Division and presented at the Annual Meeting New York, December 3 to 6, 1923, of The American Society of Mechanical Engineers.

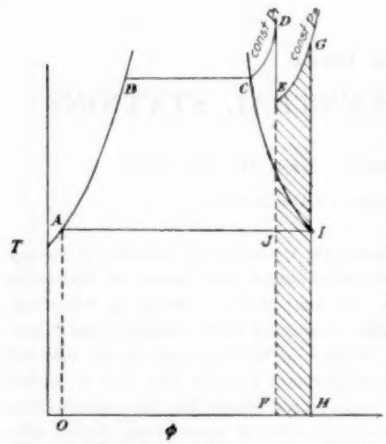

Fig. 1 One-Stage Reheating Cycle

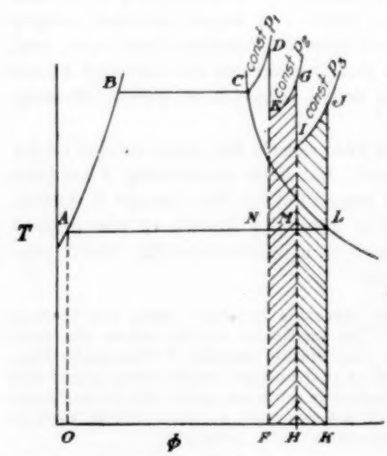

Fig. 2 Two-Stage Reheating Cycle

The Reheating Cycle

2 This cycle is illustrated on the temperature-entropy diagram in Figs. 1 and 2. In Fig. 1, *FEGH* represents the reheat area, and the gain of work is represented by *EGIJ*. In Fig. 2 the reheating is performed in the stages with an expansion *GI* between them. The gain of work is equal to *EGIJLMN*, and the energy added, to *EGIJKF*. The following questions arise:

1 Where shall reheating begin?
2 How much reheating shall be resorted to?
3 In how many stages shall reheating be accomplished?

The first question is easily answered as it is obvious that the work area *EGIJ* for a given width *JI* increases as point *E* is raised. The most efficient condition will therefore result by continuing the superheat beyond point *D*. In practice, however, available materials limit this point and the present problem is thus solved when for given limiting steam temperatures questions 2 and 3 above have been answered.

Limitations to Efficient Reheating

3 Materials place a commercial limit of about 750 deg. fahr. on the initial steam temperature, although in a few recent stations — mostly under construction — higher temperatures are to be used. These must, however, at this time be considered as of an experimental nature. In this analysis upper steam temperatures of 750 deg. fahr. will be considered as the limit of good design and 1000 deg. fahr. as a possible coming future standard.

4 With this in view the two cycles mentioned are first investigated and also a third in which three reheating stages are employed. The solution of the problem is most readily accomplished by means of the Mollier chart, and the formulas with methods used are given in Appendix No. 1.

5 Figs. 3 and 4 represent respectively the influence of number of stages and quantity of energy added on the cycle efficiency.

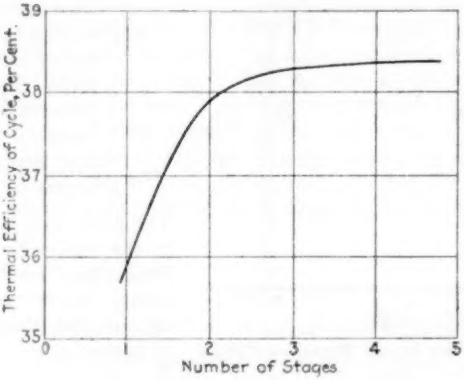

FIG. 3 INFLUENCE OF NUMBER OF STAGES IN REHEATING CYCLE
(300 lb., 750 deg. fahr., 1 in. Hg; total reheat, 300 B.t.u.)

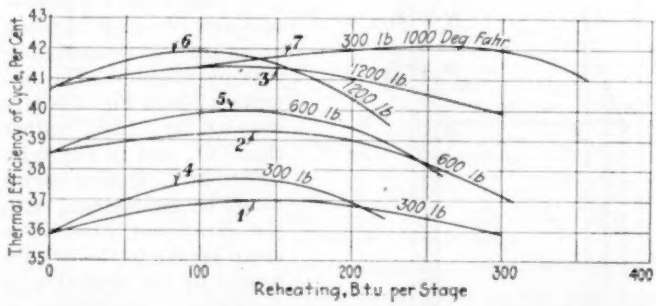

FIG. 4 INFLUENCE OF AMOUNT OF REHEATING ON CYCLE EFFICIENCY
(Reheat in each stage carried back to initial heat content of steam.)
Curves 1, 2, 3: Single-stage reheating, 750 deg. fahr. steam temperature.
Curves 4, 5, 6: Two-stage reheating, 750 deg. fahr. steam temperature.
Curve 7: Single-stage reheating, 1000 deg. fahr. steam temperature.

The following conclusions may be drawn:

a There is an appreciable gain in efficiency of the two-stage cycle over the one-stage cycle, but beyond two stages very little gain may be expected.

b For each cycle there is an optimum energy quantity per stage which varies slightly with initial pressure and considerably with initial temperature.

c The maximum efficiency increases considerably with pressure and still more with temperature. (See curve 7, Fig. 4.)

d The optimum cycle efficiencies are materially better than those of the Rankine cycle for a given initial steam temperature.

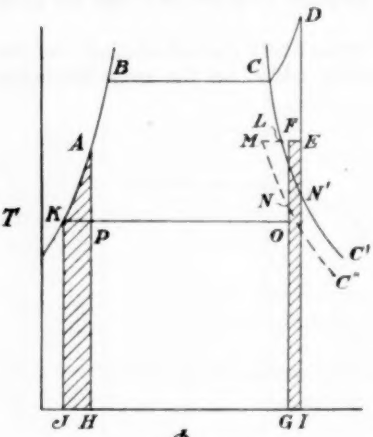

Fig. 5 One-Stage Bleeding Cycle

The latter fact is illustrated in the curves by their rise from the zero reheating point, under which conditions the Rankine cycle is performed.

Bleeding vs. Reheating Cycles

6 Figs. 5 and 6 represent one- and two-stage bleeding cycles on the temperature-entropy diagram. The saturation line in Fig. 5 is represented by $CLMC''$ rather than CC', and in Fig. 6 by $CFMGHC''$. Quality curves would be symmetrical with the saturation lines. The sharp breaks to the left are caused by the depletion in total-steam quantities at bleeding points. For every pound of steam entering the turbine only a fraction of a pound, due to bleeding, is exhausted, but the quality variations in the expansion following bleeding are just as if no bleeding had taken place. The points at which the steam condition crosses the saturation line are indicated respectively by N and N' for the bleeding and non-bleeding or Rankine cycles.

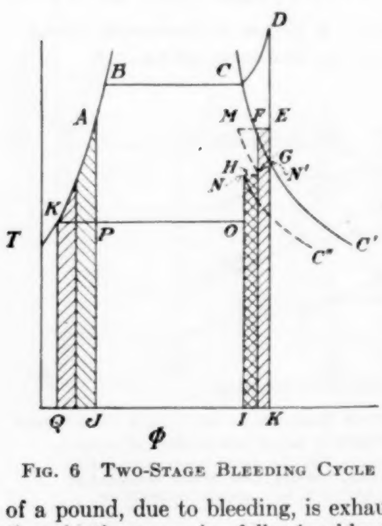

Fig. 6 Two-Stage Bleeding Cycle

7 One- and-two-stage cycles are considered, in the first of which

feedwater is heated from condensate temperature at 240 deg. fahr. and in the second, first to 180 deg. fahr. and then to 280 deg. fahr. A solution for efficiency is conveniently made with the aid of the Mollier chart, as indicated in Appendix No. 2.

8 The results are represented by Curves *D* and *E* of Fig. 7. Curves *B* and *C* show efficiencies of one- and two-stage reheating cycles for the optimum reheat quantities shown in Fig. 4. From these it is obvious that ideal bleeding cycles are more efficient than the best ideal reheating cycles. It may therefore be found advantageous to combine the reheating and bleeding processes in the same machine, and although this scheme may at first appear somewhat analogous to a tug of war between conflicting effects, such is actually not the case. The bleeding process improves the efficiency by transferring energy within the cycle, and the reheating process properly used adds energy where its availability for work is high. In addition, reheating, as later shown, is the means of materially improving the internal machine efficiency of the prime mover.

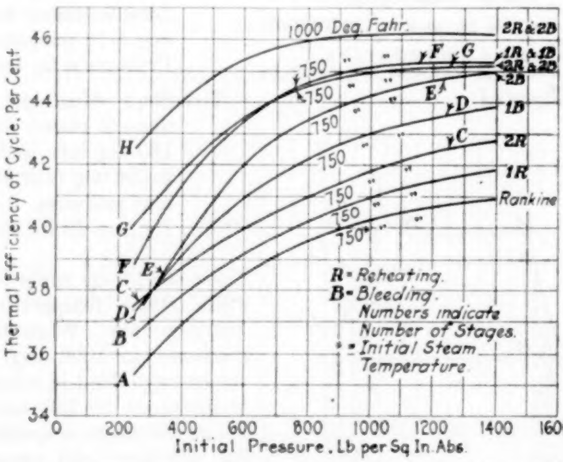

Fig. 7 Efficiencies of Various Reheating and Other Cycles

Steam conditions, *A* to *F*, inclusive: Initial temperature, 750 deg. fahr; back pressure 1 in. Hg.
Steam conditions for curve *H:* Initial temperature, 1000 deg. fahr; back pressure, 1 in. Hg.
Bleeding points: 240 deg. fahr. or 25 lb. in one-stage bleeding cycles.
280 and 180 deg. fahr. or 50 lb. and 7.5 in. Hg in two-stage cycles.

Combination Cycles

9 In Fig. 8 the *BRB*-cycle indicates one composed of two bleeding stages with reheating incorporated between them. In similar notation Kig. 9 represents the *RBRB*-cycle. Various combinations may be investigated, but from data now available the

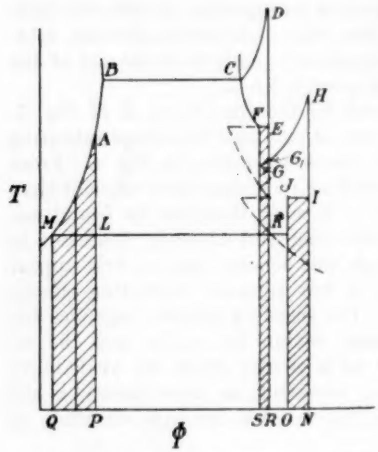

FIG. 8 *BRB*–CYCLE

Energy available for work = *ABCDEFGHIJKMA*.
Net heat added = *ABCDEG'HINPA*.
Shaded areas represent heat transferred by bleeding.
Reheat = *SGHNS*.

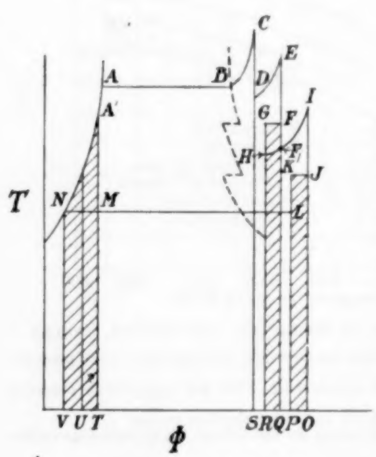

FIG. 9 *RBRB*–CYCLE

Energy available for work = *A'ABCDEFGHIJKLNA*.
Net heat added = *A'ABCDEFF'IOTA'* + *HF'QRH*.
Shaded areas represent heat transferred by bleeding.
Reheat = *SDEQ* + *RHIO*.

most favorable reheating and bleeding conditions may be chosen. They are as follows:

One-Stage Reheating: 140 B.t.u. to occur in the cycle after a 140-B.t.u. drop in the expansion line. (See Fig. 4.)

Two-Stage Reheating: 120 B.t.u. per stage occurring alternately with expansion after successive 120 - B. t. u. drops in expansion line. (See Fig. 4.)

One-Stage Bleeding: Condensate raised to 240 deg. fahr. temperature. Pressure = 25 lb. abs.

Two-Stage Bleeding: Condensate raised first to 180 deg. fahr. and then to 280 deg. fahr. Bleeding pressures, 7.5 in. Hg and 49 lb. abs.

These conditions apply for an initial steam temperature of 750 deg. fahr. When this is increased to 1000 deg. fahr. the energy added in reheating is increased to 250 and 200 B.t.u. per stage, respectively, for the one- and two-stage cycles.

10 A fulfillment of these fundamental conditions may lead to a change in order of the reheating-bleeding processes as the pressure range is followed through. This *R-B* order will, however, be left to appear as it happens to be fixed by the above more important considerations. In

solving the problems the Mollier chart again comes into play, as indicated in Appendix No. 3.

11 The resulting variations in efficiency are shown by curves *G*, *F*, and *H* in Fig. 7, from which the following conclusions may be drawn:

- *a* Combination cycles are considerably more efficient than the other cycles investigated
- *b* The ideal efficiency increases rapidly as pressures increase, but beyond 1000 lb. little is to be gained
- *c* An increase in initial temperature very materially increases the efficiency.

12 The next phase of the problem naturally has to do with relating of ideal and actual performances. This involves the influence of the various processes on the internal machine efficiencies of the prime movers.

The Internal Machine Efficiency

13 The thermal efficiency of the actual internal work will be equal to the product

$$E_c \times E_i$$

in which E_c is the thermal efficiency of the cycle and E_i the internal machine efficiency. The above product times E_m, the mechanical

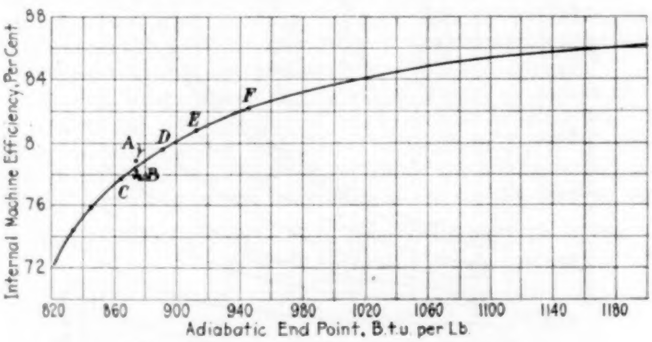

Fig. 10 Internal Machine Efficiency

A – 60,000-kw. 3-cyl. Westinghouse compound.
B – 30,000-kw. single cylinder G.E.
C, D, E, F – 30,000-kw. Westinghouse compound. Back pressure, 1 in. Hg.

efficiency, will result in the overall efficiency of work at the shaft based on the heat energy absorbed by the working medium. The efficiency product $E_i \times E_m$ involves the following losses:

- *a* Nozzle and blade frictional losses
- *b* Leakage and disk friction
- *c* Gland, bearing, and governor friction
- *d* Radiation.

The first two of these determine E_i and the fourth may be included with the third as part of the loss involved in E_m. Under these

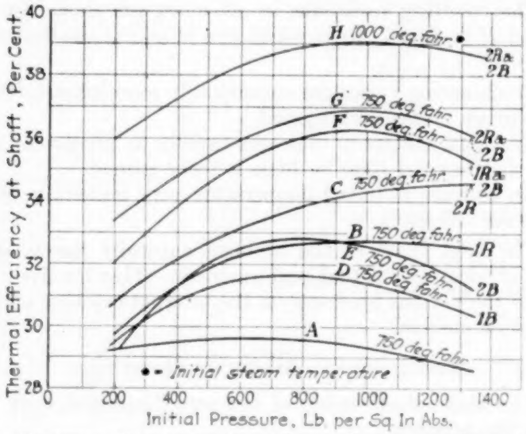

Fig. 11 Thermal Efficiencies at Shaft Based on Heat Absorbed

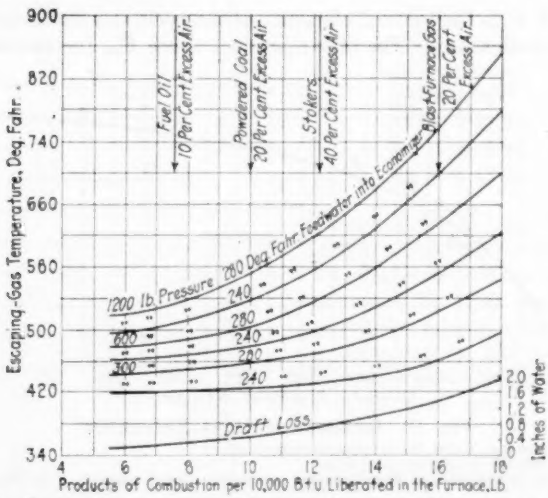

Fig. 12 Steam Generators. Relation Between Escaping-Gas Temperatures and Products of Combustion Per 10,000 B.t.u. Liberated in the Furnace

Steam temperature, 750 deg. fahr. One-third of total surface in economizer; one-pass gas flow; average rate of heat absorption, 5000 B.t.u. per sq. ft. per hr.

conditions E_m may be taken at 99 per cent without appreciable error for the cases investigated.

14 A detailed determination of E_i throughout the turbine in-

volves factors known only to the turbine designer, but its mean value may be computed from test results. These with proper corrections may be applied to the conditions of the problem. Correction factors for superheat, pressure, and vacuum have been fairly well established for the usual operating cycles, but how to apply steam-temperature corrections in reheating cycles is at first glance somewhat of a puzzle. For such cases the initial steam condition is no longer indicative of its mean condition throughout the turbine, and this, for a given turbine, is the primary factor in evaluating losses (*a*) and (*b*) above.

15 However, for a given back pressure the adiabatic end point of the expansion process is a sort of integrated value of the steam

TABLE 1 END-POINT HEAT CONTENTS AND INTERNAL MACHINE EFFICIENCIES FOR VARIOUS CYCLES AND CONDITIONS

(Heat contents in B.t.u.; efficiencies in per cent.)

Initial Pressure, lb.	Rankine and Bleeding Cycles, 750 deg. fahr. End Pt.	Mach. Effy.	One-Stage Reheating, 750 deg. fahr. End Pt.	Mach. Effy.	Two-Stage Reheating, 750 deg. fahr. End Pt.	Mach. Effy.	Two-Stage Reheating, 1000 deg. fahr. End Pt.	Mach. Effy.
300	912	80.8	983	83.4	1035	84.5	1145	85.8
600	865	77.8	940	82.0	989	83.5	1110	85.3
1200	818	71.8	890	79.6	943	82.2	1102	85.2

condition throughout, particularly if the reheating has occurred, as it should have, early in the cycle. Such end points can always be found by tracing out the ideal cycle on the Mollier chart. If for a machine under consideration test data are available, actual internal machine efficiencies may be plotted against adiabatic end points for a given back pressure.

16 Fig. 10 represents such a curve plotted by means of test data available for a number of large turbines as indicated. The data and computations are included in Appendix No. 4. To obtain the internal machine efficiency, find first the adiabatic end point of the cycle in question by tracing it out on the Mollier chart.

TABLE 2 REHEATER PRESSURES FOR VARIOUS BOILER PRESSURES

(Pressures in lb. per sq. in.)

Initial Pressure	Reheater Pressures — One Stage	Two Stages — 1st Stage	2d Stage
300	88	105	46
600	170	210	70
1200	290	380	120

Then find the efficiency value corresponding to this condition from the curve. Such curves have the advantage of automatically correcting for superheat, initial pressure, and reheat. The limitations to their use are outlined in Appendix No. 4.

Thermal Efficiency at Shaft Based on Heat Absorbed

17 The end-point heat contents and internal machine efficiencies for the cycles and conditions considered are given in Table 1. Reference to this table brings out the following conditions:

a Unless reheating is used the machine efficiency is relatively low at high pressures

b With 1000 deg. fahr. steam temperature the machine efficiency is 3 per cent better for two-stage reheating at 1200 lb. pressure than it is with 750 deg. fahr. steam temperature.

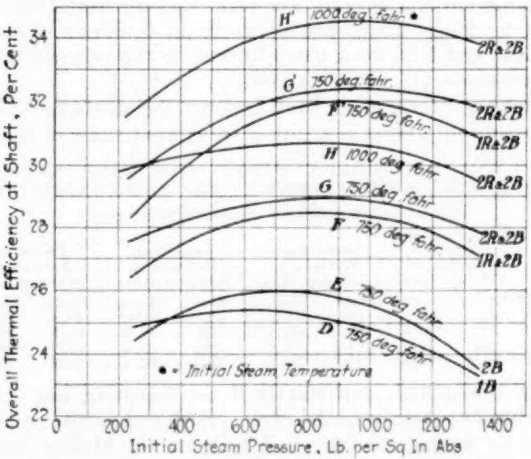

FIG. 13 OVERALL THERMAL EFFICIENCIES OF A GIVEN STEAM-GENERATING PLANT FOR VARIOUS COMBINATION AND BLEEDING CYCLES

(Auxiliaries, electric generator and pipe-line radiation losses not included.)

Curves *D*, *E*, *F*, *G*, and *H* for a stoker-fired plant.
Curves *F'*, *G'*, and *H'* for a powdered-coal maximum-efficiency plant in which steam-generator efficiency is 88 per cent.

18 The thermal efficiencies at the shaft based on heat absorbed may now be obtained and are represented in Fig. 11. The following conditions are apparent:

a Because of its effect on machine efficiency, reheating becomes more efficient than bleeding

b The highest efficiency is obtained in the two-stage combination cycle, i.e., 2*R* and 2*B*

c All curves shown have maximum points and for 750-deg. fahr. steam these are in most cases below 1000 lb. pressure.

19 The fact that steam generators naturally fall off in efficiency with increasing pressures will tend to throw the high-efficiency points still farther down the pressure scale.

THE STEAM GENERATOR IN REHEATING STATIONS

20 Reheating surface will normally be included within the boiler setting to withdraw heat from the products of combustion. Mean steam temperatures within them will be somewhat higher

than boiler-surface temperatures for 300-lb. and 600-lb. pressures, but for 1200-lb. stations they will actually be somewhat lower. The pressures existing within the reheaters will be considerably below those existing in the boilers, as shown in Table 2. It appears, therefore, that for a given heat absorption, reheater surface advantageously placed [1] may not be more expensive than boiler surface. This would be particularly true if the reheating surface should be exposed directly to furnace radiation, as under these conditions the rate of heat absorption would be very high. Such an arrangement also has obvious advantages when considering maintenance of furnace walls and should lead to higher steam-generator efficiencies because a larger proportion of the total liberated heat energy may be absorbed in the furnace, leaving therefore less energy to be absorbed by convection. The reheater maintenance may, however, be high and there are some operating disadvantages which may make the arrangement undesirable particularly at light loads when the reheating would be excessive.

TABLE 3 STEAM-GENERATOR EFFICIENCIES FOR VARIOUS CYCLES AND PRESSURES

	Steam pressure, lb. per sq. in.	300	600	1200
One-Stage Bleeding	Escaping-gas temperature, deg. fahr.	430	500	580
	Combined efficiency, per cent	82.8	80.8	78.4
Two-Stage Bleeding	Escaping-gas temperature, deg. fahr.	470	540	620
	Combined efficiency, per cent	81.8	79.8	77.8
Rankine	Efficiency, per cent (90 deg. fahr. feedwater)	85.2	83.2	80.8

21 Under variable-load operation reheaters screened from the furnace will function much as superheaters in the gas-convection

[1] See Appendix No. 1.

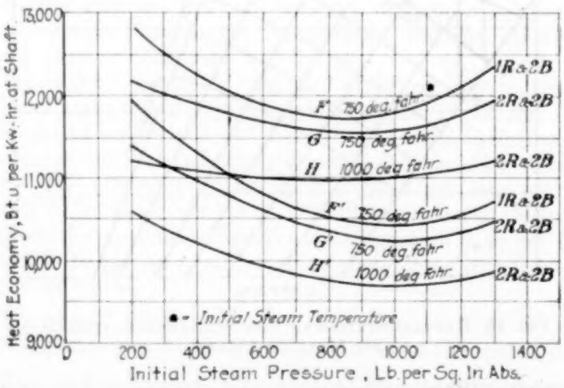

FIG. 14 OVERALL ECONOMIES FOR STATIONS EMPLOYING VARIOUS COMBINATION CYCLES

(Auxiliaries, electric generator and pipe-line radiation losses not included.)

Curves *F*, *G*, and *H* for stoker-fired plants.
Curves *F'*, *G'*, and *H'* for high-efficiency powdered-coal plants in which steam-generator efficiency is 88 per cent.

path. If the steam-generator driving rate is maintained proportional to the load on the prime mover, little difficulty should be experienced at light loads, as the ratio of gas weight to steam weight will remain substantially constant. For these reasons the reheater in the gas-convection path appears to have, in the present state of the art, more desirable operating characteristics. When the heat absorption in reheaters is large it may, however, be found advantageous to absorb part of the energy directly from furnace radiation.

Steam-Generator Efficiencies

22 For high pressures boiler-surface temperatures will be increased with a corresponding rise in the escaping-gas temperature.

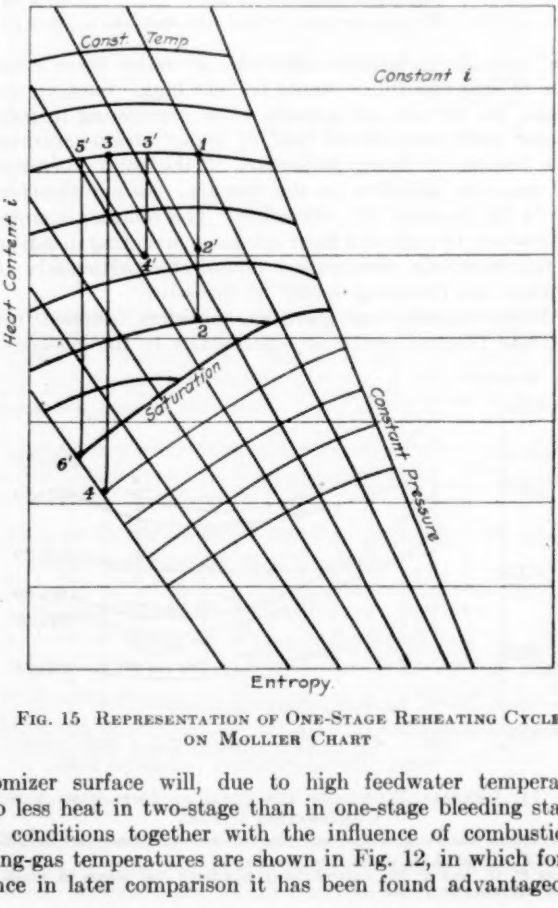

Fig. 15 Representation of One-Stage Reheating Cycle on Mollier Chart

Economizer surface will, due to high feedwater temperatures, absorb less heat in two-stage than in one-stage bleeding stations. These conditions together with the influence of combustion on escaping-gas temperatures are shown in Fig. 12, in which for convenience in later comparison it has been found advantageous to

state the rate of driving in terms of the average rate of heat absorption in B.t.u. per sq. ft.

23 The expected efficiencies for a stoker-fired plant with a steam generator proportioned as shown and absorbing heat at the average rate of 5000 B.t.u. per sq. ft. per hr. are stated in Table 3.

Overall Efficiencies for a Given Steam-Generating Plant

24 Curves *D*, *E*, *F*, *G*, and *H*, Fig. 13, represent the overall efficiencies by combining the foregoing steam-generator efficiencies with those from Fig. 11. The economies in B.t.u. per kw-hr. for certain of these stations are shown by curves *F*, *G*, and *H* of Fig. 14. It is apparent from these curves that no gain may be expected by increasing pressures beyond 850 lb.

Stations of Maximum Efficiency

25 When high load factor and cost of fuel make feasible the use of steam generators of exceedingly high efficiencies, the following modifications may be made:

a Decreased average rate of heat absorption
b Increased ratio of economizer to boiler surface
c Installation of air preheaters
d Burning coal in powdered form.

Under these conditions average steam-generator efficiencies of 86 to 90 per cent may be realized even for the high-pressure two-stage-bleeding stations, but to accomplish efficiencies appreciably higher than this even in the low-pressure stations would undoubtedly involve prohibitive conditions.

26 For a steam-generator efficiency of 88 per cent a new set of efficiency and economy curves represented by F', G' and H' in Figs. 13 and 14 are arrived at. Reference to them indicates that for 750-deg. steam the maximum-efficiency point occurs at 1000 lb. pressure. The efficiency has a value of 32.3 per cent and the economy is 10,300 B.t.u. per kw-hr. at the turbine shaft. Generator losses, power for auxiliaries, and pipe-line radiation have not been included. If 10 per cent is charged to these three items the net results are as follows:

Net overall efficiency = 29.1 per cent
Net economy = 11,500 B.t.u. per kw-hr. at the switchboard.

27 We may conclude that the station with a net overall economy of 12,000 B.t.u. is in sight. Its characteristics and principal dimensions may be about as follows:

Initial steam pressure, 900 to 950 lb. per sq. in.
Initial steam temperature, 750 deg. fahr.
Back pressure, 1 in. Hg.

Steam Cycle: Combination $2R + 2B$.
Reheating, 120 B.t.u. per stage per lb.
Bleeding, 49 lb. and 7.5 in. Hg.
Condensate, 280 deg. fahr. and 180 deg. fahr.

Steam Generator:
Surface equally distributed between boiler and economizer.
Average rate of heat absorption, 3000 B.t.u. per sq. ft. per hr.
Air-preheater surface per sq. ft. of surface in boiler and economizer, 0.2 sq. ft.

Boiler, superheater, and economizer surface per kw. $= \dfrac{12{,}000}{3000} \times 0.88 = 3.52$ sq. ft.

Boiler, superheater, and economizer surface corrected for reheating per kw. $= \dfrac{1345 - 248}{1345 - 248 + 240} \times 3.52 = 2.89$ sq. ft.

Reheating surface per kw. = (?) depends on position
Energy absorbed in reheater per kw-hr. = 1900 B.t.u. = 18 per cent of total

Air-preheater surface per kw. $= 0.2 \times 3.52 = 0.70$ sq. ft.
Temperature of gases escaping from boiler, 620 deg. fahr.
Temperature of gases escaping from economizer, 350 deg. fahr.
Temperature of gases escaping from air preheater, 260 deg. fahr.
Draft loss in boiler and economizer, 1 in. water.
Draft loss in air preheater, 4 to 5 in. water

Fuel: Powdered coal burned with 20 per cent excess air.

Comparison of Stations

28 A brief comparison of several station types may now be of interest.

Station No. I — Steam pressure, 700 lb. per sq. in.
Steam temperature, 750 deg. fahr.
Back pressure, 1 in. Hg.

Cycle: Combination $2R + 2B$.
Fuel: Bituminous coal burned on stokers with 40 per cent excess air.
Economy per kw-hr. at shaft, 11,650 B.t.u. (See *G*, Fig. 14.)
Correction for auxiliaries, electric generator and pipe-line losses, 8 per cent.

Net economy $= \dfrac{11{,}650}{0.92} = 12{,}700$ B.t.u.

Surface in boiler, superheater, and economizer per net kw.

$$= \frac{12{,}700}{5000} \times 0.80 = 2.03 \text{ sq. ft.}$$

Surface in boiler, superheater, and economizer per kw. corrected for reheating $= 0.824 \times 2.03 = 1.67$ sq. ft.
Energy absorbed per kw-hr. in reheaters = 1790 B.t.u. = 17.7 per cent of total.
Economizer surface per kw. $= 0.33 \times 2.03 = 0.67$ sq. ft.
Draft loss in boiler and economizer, 0.8 in. water (one-pass gas flow).

Station No. II — Steam pressure, 900 lb. per sq. in.
Steam temperature, 750 deg. fahr.
Back pressure, 1 in. Hg.

Cycle: Combination $2R + 2B$.
Fuel: Powdered coal burned with 20 per cent excess air.
Economy per kw-hr. at shaft, 10,300 B.t.u. (See *G*, Fig. 14.)
Correction for auxiliaries, electric generator and pipe losses, 12 per cent.

Net economy per kw-hr. $= \frac{10{,}300}{0.88} = 11{,}700$ B.t.u.

Surface in steam-generating plant:

Boiler + superheater + economizer $= \frac{11{,}700}{3000} \times 0.88 = 3.43$ sq. ft. per kw.

Surface in boiler, superheater, and economizer per kw. corrected for reheating $= 0.822 \times 3.43 = 2.8$ sq. ft.

Energy absorbed in reheaters per kw-hr. = 1850 B.t.u. = 18.2 per cent of total.

Economizer surface per kw. $= 0.50 \times 3.43 = 1.72$ sq. ft.

Air preheater $= 0.2 \times 3.43 = 0.69$ sq. ft. per kw.

Draft loss, boiler + economizer + air preheater = 6 in. water.

Station No. III — Steam pressure, 1000 lb. per sq. in.
Steam temperature, 1000 deg. fahr.
Back pressure, 1 in. Hg.

Cycle: Combination $2R + 2B$.
Gross economy at shaft, 9700 B.t.u. (See *H'*, Fig. 14.)

Net economy $= \frac{9700}{0.88} = 11{,}000$ B.t.u. per kw-hr.

Surface in steam-generating plant:

Boiler, superheater, and economizer $= \frac{11{,}000}{3000} \times 0.88 = 3.23$ sq. ft. per net kw.

Surface in boiler, superheater, and economizer corrected for reheating per kw. $= 0.755 \times 3.23 = 2.44$ sq. ft.

Energy absorbed in reheaters per kw-hr. = 2370 B.t.u. = 24.5 per cent of total

Economizer surface per kw. $= 0.50 \times 3.23 = 1.62$ sq. ft.

Air preheater $= 0.20 \times 3.23 = 0.65$ sq. ft. per net kw.

Fuel: Powdered coal burned with 20 per cent excess air.
Draft loss, 6 in. water.

29 Inspection shows that station No. II as compared to No. I has an 8 per cent lower fuel consumption but requires 68 per cent more surface in boilers and economizers and also 34 per cent additional surface in air preheaters. The pressure for station No. II is nearly 30 per cent higher and powdered-coal equipment is used as against stokers in station I. Station II must therefore be charged with the extra cost of:

a One and seven-tenths times the surface at 30 per cent higher pressure
b An additional 34 per cent in air preheaters
c Increased draft auxiliaries
d Extra cost of powdered-coal equipment
e Additional surface in reheaters.

However, for variable-load operation decreased stand-by losses and better combustion control in the powdered-fuel station may

lead to an actual fuel saving over a period of time several per cent greater than the above figure of 8 per cent, and before coming to any conclusion these important factors should of course be given their due consideration. Station No. III as against No. II saves about 6.3 per cent of the fuel with about 13 per cent less surface, excluding the reheaters, at 10 per cent higher pressure but at an initial steam temperature so much higher that the extra cost of maintenance and investment would make its use of doubtful value at the present time. Furthermore station No. III will require considerably more reheating surface, as is shown by the figure of 2370 B.t.u. per kw-hr., as against 1850 B.t.u. per kw-hr. for station No. II.

Conclusions

30 The following conclusions may be drawn from what has been presented:

a With increasing pressures and a limited steam temperature reheating is becoming essential for highest efficiencies.

b The introduction of reheating stages does not operate to make bleeding stages less beneficial, and in the future stations both bleeding and reheating stages will probably be incorporated in the steam cycle.

c Under present limiting conditions as to steam temperature there is little to be gained by using more than two reheating stages.

d Reheating should be accomplished as early in the expansion period as possible.

e For variable-load operation the reheating coils should preferably be placed in the gas-convection path screened from direct furnace radiation and the load on the steam generators maintained proportional to that of the prime movers.

f For stoker-fired plants using the combination cycle material gains in economy should result by increasing pressures up to between 700 and 800 lb., but beyond this point no apparent gains exist.

g In powdered-fuel plants using the combination cycle material gains in economy may be expected by increasing pressures up to 900 lb., but beyond this point no apparent gains exist.

h To increase pressures materially above points noted in (*b*) and (*g*) will apparently result in an actual reduction of economy for the conditions stated.

i For properly designed stoker-fired plants operating on the combination cycle at optimum conditions, economies of 13,000 B.t.u. are apparently in sight.

j For properly designed powdered-fuel plants operating under optimum conditions, economies of 12,000 B.t.u. and better appear possible.

k Future improvements in turbine design may slightly influence optimum pressures. Such improvements will, however, probably

lead to a proportional shift in the internal-machine-efficiency curve, resulting merely in a higher efficiency at nearly the same optimum pressure.

l More accurate information on the properties of high-pressure steam may lead to a readjustment of optimum pressures for the conditions stated. It is thought, however, that only small readjustments will result from this cause.

APPENDIX NO. 1

THE REHEATING CYCLE

31 Referring to Fig. 15, the one-stage reheating cycle is represented by 1 2 3 4.

Adiabatic work, 1 to 2 = $i_1 - i_2$.
Reheating at constant pressure, 2 to 3 = $i_3 - i_2$.
Adiabatic work, 3 to 4 = $i_3 - i_4$.
Total work = $(i_1 - i_2) + (i_3 - i_4)$.
Total heat absorbed = $(i_1 - i_4) + (i_3 - i_2)$.

$$\text{Thermal efficiency} = \frac{(i_1 - i_2) + (i_3 - i_4)}{(i_1 - i_4) + (i_3 - i_2)}.$$

32 Similarly, for the two-stage cycle (12′3′4′5′6′),
Work = $(i_1 - i_2') + (i_3' - i_4') + (i_5' - i_6')$.
Heat absorbed = $(i_1 - i_6') + (i_3' - i_2') + (i_5' - i_4')$.

$$\text{Thermal efficiency} = \frac{\text{Work}}{\text{Heat absorbed}}.$$

33 It is to be noted that reheating in this analysis is carried to the initial steam heat content rather than to the initial steam temperature in spite of the fact that fulfillment of the latter conditions will result in a slightly higher optimum thermal efficiency of the steam cycle. By the former method steam temperatures in successive reheating stages are somewhat reduced, and that this condition is necessary if reasonable surface proportions are to be realized in two-stage reheating screened from direct radiation is apparent from the following statements.

34 Assuming a stoker-fired plant at 83 per cent heat absorption, efficiency based on liberated heat energy, the ratio of gas to steam weights will be about as follows: For 600 lb. pressure, 750 deg. fahr. steam, 280 deg. fahr. feedwater, and reheating 240 B.t.u. in 2 stages:

Products of combustion per 10,000 B.t.u. liberated = 12.2 lb.
Heat energy absorbed per 10,000 B.t.u. liberated = 8300 B.t.u.
Lb. of steam generated per 10,000 B.t.u. liberated

$$= \frac{8300}{(1371 - 248) + 240} = 6.08 \text{ lb.}$$

$$\text{Lb. gas per lb. steam} = \frac{12.2}{6.08} = 2.$$

35 The gas-temperature drop per B.t.u. absorbed by steam will be nearly equal to $1/(0.25 \times 2) = 2$ deg. fahr.

36 If the superheater and reheaters are to be screened from furnace radiation by boiler heating surface, the temperature of gases entering the superheating surface will probably not be over 1400 to 1500 deg. fahr.

The superheater absorbs 178 B.t.u. per lb. of steam. Hence the gas-temperature at entrance to the first reheater may easily be as low as [1400 $-(2 \times 178) =$] 1044 deg. fahr., and at entrance to the second reheater it will be reduced to [1044 $-$ $(2 \times 120) =$] 794 deg. fahr.

37 This would result in a temperature differential of less than 50 deg. fahr. between gas and steam at entrance to the second reheater, assuming counterflow. If the reheat is carried to the initial heat content only this differential is increased to over twice this value; i.e., 110 deg. fahr., and hence the surface requirements in the second reheater would be very materially reduced.

38 For powdered fuel the surface requirements would be even greater because of the reduced gas weights, as with 20 per cent excess air the gas-temperature drop per B.t.u. absorbed in the reheater might easily increase to 2.5 deg. fahr. as against 2 deg. fahr. for the above case, and, if the initial steam temperature is 1000 deg. fahr., it would undoubtedly require reheaters exposed to furnace radiation for optimum reheats in a two-stage reheating cycle, whatever the fuel.

39 For one-stage reheating much less heat is absorbed and hence temperatures could for most conditions probably be carried up to initial values with little difficulty.

APPENDIX NO. 2

BLEEDING CYCLES

40 *One-Stage Cycle* (Fig. 16):

Heat in bled steam per lb. $= [i_1 - (i_1 - i_2)] = i_2$.

$(i_1 - i_2) =$ adiabatic heat drop on Mollier chart from p_1, t_1 to p_2.

Quantity of bled steam per lb. condensate:

$$w_2[i_2 - (t_2 - 32)] = 1[t_2 - t_n].$$

Solye for w_2.

Ideal work by 1 lb. condensate

$= (i_1 - i_n)$ by adiabatic drop from p_1, t_1 to p_n.

Ideal work by w_2 lb. bled steam $= w_2(i_1 - i_2)$.

Total work by $(1 + w_2)$ lb. steam $= (i_1 - i_n) + w_2(i_1 - i_2)$.

Total heat absorbed by $(1 + w_2)$ lb. steam

$$= (1 + w_2)\,[i_1 - (t_2 - 32)].$$

Thermal efficiency of cycle

$$= \frac{(i_1 - i_n) + w_2(i_1 - i_2)}{(1 + w_2)\,[i_1 - (t_2 - 32)]}.$$

41 *Two-Stage Cycle* (Fig. 17):

Quantity w_3 of bled steam:

$$w_3[i_3 - (t_3 - 32)] = 1[t_3 - t_n].$$

Solve for w_3. i_3 obtained from relation

$$i_3 = [i_1 - (i_1 - i_3)]$$

in which $(i_1 - i_3) =$ adiabatic drop from p_1, t_1 to p_3.

In similar manner solve for w_2 from relation

$$w_2[i_2 - (t_2 - 32)] = (1 + w_3)\,[t_2 - t_3].$$

i_2 obtained by adiabatic drop to p_2 from p_1, t_1 from relation

$$i_2 = [i_1 - (i_1 - i_2)].$$

Total work by $(1 + w_2 + w_3)$ lb. steam

$$= (i_1 - i_n) + w_2(i_1 - i_2) + w_3(i_1 - i_3).$$

Heat absorbed by $(1 + w_2 + w_3)$ lb. steam

$$= (1 + w_2 + w_3)\,[i_1 - (t_2 - 32)].$$

Thermal efficiency of cycle

$$= \frac{(i_1 - i_n) + w_2(i_1 - i_2) + w_3(i_1 - i_3)}{(1 + w_2 + w_3)\,[i_1 - (t_2 - 32)]}.$$

APPENDIX NO. 3

COMBINATION CYCLES

42 *One Stage Reheating, Two Stages Bleeding.* Referring to Fig. 18, the *RBB*-cycle is represented on the Mollier chart by path *abcn*. On this path (2) and (3) represent bleeding points and *ba* the reheating stage at constant pressure to initial heat content after adiabatic expansion from

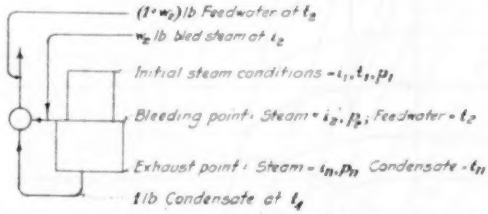

FIG. 16 DIAGRAM OF ONE-STAGE BLEEDING CYCLE

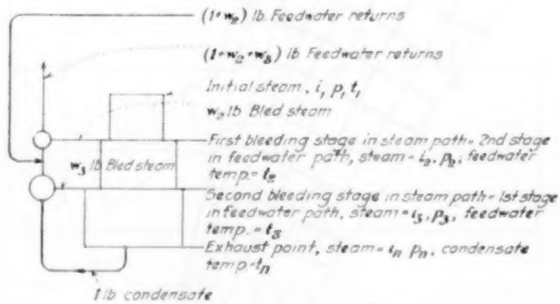

FIG. 17 DIAGRAM OF TWO-STAGE BLEEDING CYCLE

initial condition to point *b*. The quantities of bled steam are solved for from condition points and temperatures just as in the bleeding cycle. The following relations will be found true:

Work by 1 lb. steam through turbine $= (i_a - i_b) + (i_c - i_n) = W_1$
Work by w_2 lb. bled steam $= w_2[(i_a - i_b) + (i_c - i_2)] = W_2$
Work by w_3 lb. bled steam $= w_3[(i_a - i_b) + (i_c - i_3)] = W_3$
Work by $(1 + w_2 + w_3)$ lb. steam $= W_1 + W_2 + W_3$
Heat absorbed by $(1 + w_2 + w_3)$ lb. steam

$$= (1 + w_2 + w_3)\{[i_a - (t_2 - 32)] + (i_c - i_b)\}.$$

Thermal efficiency of cycle

$$= \frac{W_1 + W_2 + W_3}{(1 + w_2 + w_3)\{[i_a - (t_2 - 32)] + (i_c - i_b)\}}.$$

43 *Two Stages Reheating, Two Stages Bleeding.* The *RBRB*-cycle is represented on the Mollier chart (Fig. 19) by the path *abcden*. Bleeding points are shown at (2) and (3), and reheating stages at constant pressure from *b* to *c* and from *d* to *e* after successive adiabatic expansions first from *a* to *b* and then from *c* to *d*. The quantities of bled steam are solved for as before from the condition points and temperatures involved.

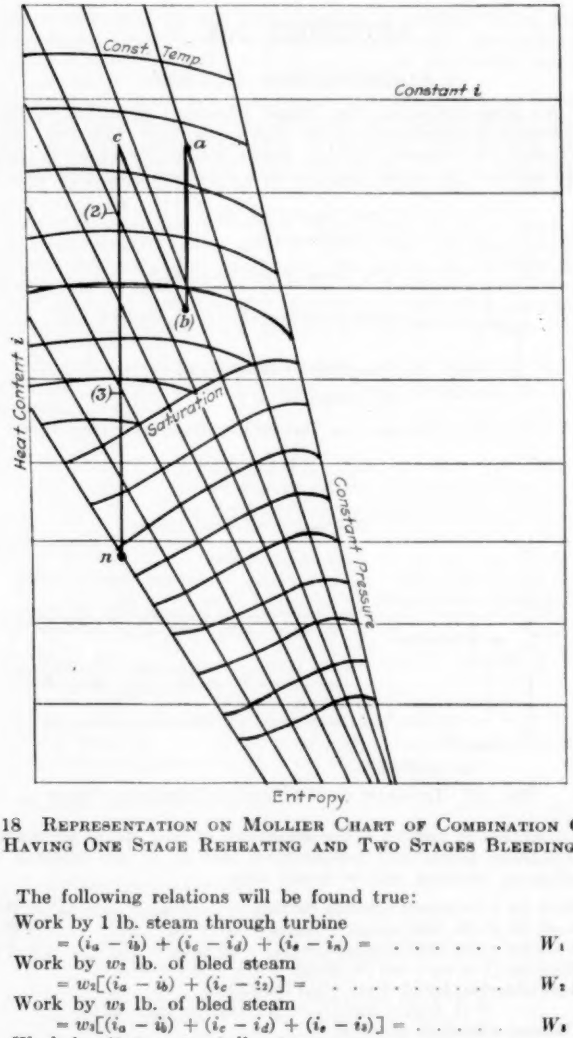

FIG. 18 REPRESENTATION ON MOLLIER CHART OF COMBINATION CYCLE HAVING ONE STAGE REHEATING AND TWO STAGES BLEEDING

44 The following relations will be found true:

Work by 1 lb. steam through turbine

$= (i_a - i_b) + (i_c - i_d) + (i_e - i_n) =$ . . . . W_1

Work by w_2 lb. of bled steam

$= w_2[(i_a - i_b) + (i_c - i_2)] =$ . . . . W_2

Work by w_3 lb. of bled steam

$= w_3[(i_a - i_b) + (i_c - i_d) + (i_e - i_3)] =$ . . . . . . W_3

Work by $(1 + w_2 + w_3)$ lb. steam

$= W_1 + W_2 + W_3.$

Heat absorbed by $(1 + w_3)$ lb. steam

$= (1 + w_3) \{[i_a - (t_3 - 32)] + (i_c - i_b) + (i_e - i_d)\} = Q_{13}$

Heat absorbed by w_2 lb. steam

$= w_2 \{[i_a - (t_2 - 32)] + (i_c - i_b)\} =$ . . . . Q_2

Thermal efficiency of cycle

$$= \frac{W_1 + W_2 + W_3}{Q_{13} + Q_2}.$$

45 It is now apparent that in the solution of such problems it is necessary to find for each of the steam quantities, w_2, w_3 and the 1 lb. through the turbine, (1) the work done and (2) the heat absorbed. The total work accomplished is the sum of the separate work quantities and the total heat absorbed the sum of the separately absorbed heat-energy quantities. The ratio of summation of work divided by summation of absorbed heat-

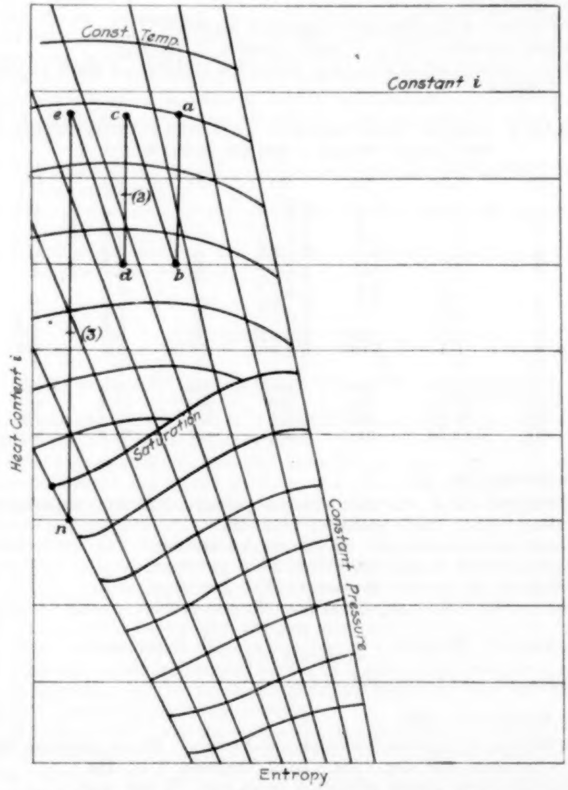

Fig. 19 Representation on Mollier Chart of Combination Cycle Having Two Stages Reheating and Two Stages Bleeding

energy quantities is equal to the thermal efficiency of the cycle. In solving the problem care must be taken in determining just exactly how much work is done, and heat absorbed, by each of the steam quantities involved.

APPENDIX NO. 4

RELATION BETWEEN INTERNAL MACHINE EFFICIENCY AND ADIABATIC END POINT

PRIME MOVER No. I:

30,000-kw. Westinghouse compound machine.
Steam pressure, 215 lb.; back pressure, 1 in. Hg.
Steam consumption and heat converted to internal work as given in Table 4.

TABLE 4 STEAM CONSUMPTION AND HEAT CONVERTED TO INTERNAL WORK — PRIME MOVER NO. I

(1) Superheat, deg. fahr.	(2) Corrected steam consumption	(3) Heat converted to work per lb. steam, B.t.u.	(4) Corrected for bearing, gland, governor and generator loss [col (3) × 1.065]	(5) Adiabatic drop	(6) Internal machine efficiency, per cent	(7) Adiabatic end point, B.t.u.
50	12.36	276	294	388	75.8	846
100	11.75	290	309	398	77.6	863
150	11.25	303	323	412	78.5	876
200	10.80	316	337	424	79.6	889

PRIME MOVER No. II:

30,000-kw. G. E. turbine. Steam pressure, 225 lb.; superheat, 150 deg. fahr.; back pressure, 1 in. Hg.
Steam consumption, 11.03 lb. per kw-hr. net.
Rankine-cycle engine efficiency, 75.5 per cent.
Efficiency of electric generator, 97.5 per cent.
Mechanical efficiency of turbine, 99 per cent.

$$\therefore \text{ internal efficiency} = \frac{0.755}{0.99 \times 0.975} = 78.3 \text{ per cent.}$$

Adiabatic end point, 875 B.t.u.

PRIME MOVER No. III:

60,000-kw. 3-cylinder Westinghouse turbine. Steam pressure, 220 lb. superheat, 150 deg. fahr.; back pressure, 1 in. Hg.
Rankine-cycle engine efficiency from test, 76 per cent.

$$\text{Internal machine efficiency} = \frac{0.76}{0.965} = 78.8 \text{ per cent.}$$

Adiabatic end point, 876 B.t.u.

46 The curve represented in Fig. 10 is plotted from data given in columns 6 and 7 of Table 4. It is to be noted that computed points for prime movers II and III fall just below and above this curve, indicating that each machine will have its own curve. The deviation is so slight, however, that possible errors in assumptions if corrected might bring all three prime movers to the same point at the same end condition for the same back pressure. It is to be expected, however, that small differences will exist for machines of different types of practically the same output. The important thing is the trend of the curve, and this it is felt is very

nearly correct for the machine considered. Machines A and B might produce curves with a somewhat different trend, but probably the difference would be very small.

47 Such a relation, however, must be used with judgment as it is possible to effect materially the end point without drawing a corresponding influence on the mean condition of the steam throughout the cycle, as, for instance, by reheating the steam just before it is discharged. This process is of course exceedingly inefficient and will decrease the cycle efficiency itself. In general, therefore, we may say that the relationship is reasonably correct for a given machine at a given back pressure when the reheating is performed, as it should be, early in the cycle. In other words, when the cycle efficiency is favorably affected by reheating then the machine efficiency is likewise bound to be favorably affected. The following additional conditions should be kept in mind in using such a relationship:

a The relation will be different for small machines from what it is for large machines

b The type of the machine, especially in small sizes, will influence the relationship

c For a given machine size and type the relation will be influenced by the back pressure.

APPENDIX NO. 5

SURFACE REQUIREMENTS IN THE REHEATERS

48 When energy as heat is transmitted from gases through a metal wall to a liquid in contact with the other side of the wall the mean temperature difference is easily determined because the wall temperature will be practically that of the liquid which is absorbing the energy. For superheaters, reheaters and air preheaters, however, the wall temperature due now to the existence of surface films on both sides, will be between that of the heat-absorbing and heat-rejecting medium. Before surface calculations may be made it is necessary to determine the probable wall temperature.

49 These wall or surface temperatures will depend upon the relation between the rates of heat transfer on the two sides. For boilers and economizers this ratio is very high; that is, the rate from surface to water is many times that from hot gases to surface.

50 This rate of heat transfer for a given gas and surface arrangement depends primarily on the mean temperature difference and gas velocity. It varies also with the nature of the gas.

51 A little thought will show that the wall temperatures may be varied at will by a difference in relation of gas velocities on the two sides of the wall and the higher the mean wall temperature the more surface will be required for a given heat absorption. *Reheaters should therefore be designed for high steam velocities in order not only to decrease the surface requirements but also to decrease the cost of maintaining what surface is installed.* For infinite steam velocities the heat transfer from gas to surface will be equal to that in boilers having the same difference in temperature between the hot and cold media. For any finite steam velocities the heat transfer must therefore be less.

52 By equating the energy absorbed to that given up the mean wall temperature may be approximated as follows:

$$K_g F(t_g - t_w) = K_s F(t_w - t_s)$$

in which

F = wall surface,

K_g and K_s = coefficients of heat transfer, gas to wall and wall to steam

t_g, t_s and t_w = respectively mean temperatures of gas, steam and wall.

Solving for t_w

$$t_w = \frac{t_g + \frac{K_s}{K_g} \times t_s}{\frac{K_s}{K_g} + 1}$$

53 To apply this relation, the coefficients K_s and K_g must be evaluated. For boiler and economizer surfaces there are fairly good data available for determining the coefficient K_g. At least an approximation and the trend of variation of K_s may be found from the relation developed by Nusselt, and the conditions of his experiments are near those involved for convection-type reheaters used in Stations I and II. The temperatures involved for Station III are somewhat beyond the range of Nusselt's experiments, but extrapolation may result in a fair estimate of what might happen.

54 Nusselt's relation is as follows:

$$K_s = 15.90 \frac{\lambda_{wall}}{d^{0.214}} \left(\frac{wpc_p}{\lambda BT}\right)^{0.786}$$

in which

λ_{wall} = coefficient of conduction of gases at wall temperature
d = diameter of tube, ft.
w = velocity of gas, ft. per sec.
c_p = specific heat at constant pressure, B.t.u. per lb.
λ = coefficient of conduction of gas at temperature in tube
p = pressure of gas in tube, lb. per sq. ft.
B = equivalent gas constant = $P_m V_m / T_m$
P_m = mean pressure, lb. per sq. ft.
V_m = mean specific volume, cu. ft. per lb.
T_m = mean absolute temperature of gas in tube.

For superheated steam

$$\lambda = 0.01288 \left[1 + 241 \times 10^{-5}(t - 32)\right]$$

55 *Application to Conditions of Station 1.*

	First stage	*Second stage*
Reheating pressure, lb. per sq. in.	240	80
Temperature of steam at entrance to reheater, deg. fahr.	470	430
Temperature of steam leaving reheater, deg. fahr.	690	677
Temperature of gases at entrance to reheater, deg. fahr.	1044	794
Temperature of gases leaving reheater, deg. fahr.	794	554
Mean steam temperature in reheater, deg. fahr.	580 = 1040 deg. fahr. abs.	553 = 1013 deg. fahr. abs.
Mean gas temperature in reheater, deg. fahr.	919 = 1379 deg. fahr. abs.	674 = 1134 deg. fahr. abs.

56 Applying the conditions in the first reheater to Nusselt's formula, the following values are obtained:

$$\lambda_{wall} = 0.0306 \qquad \lambda_{steam} = 0.0299$$
$$B = 82.8 \qquad C_p = 0.545$$
$$P = 34{,}600 \qquad T = 1040$$

For a 2-in. tube, the following values of K_s are obtained:

Steam velocity, ft. per sec.	50.0	100.0	200.0
K_s = ...	12.3	21.0	29.8

57 We shall now arrange the reheating surface so that the mean coefficient K_g (gas to wall) is 7. For the condition stated, this would require a gas velocity of about 1500 ft. per min. Substituting in the foregoing relation for t_w, the mean wall temperatures are found to be as follows:

Steam velocity, ft. per sec.	50	100	200
Mean wall temperature t_w, deg. fahr.	702	664	644
Mean temperature difference, gas and wall, deg. fahr. .	217	255	275

58 *Surface in First-Stage Reheater per Kw.* From former data, the B.t.u. per kw-hr. absorbed in reheating for Station I = 10,150 × 0.175 = 1780. One half of this must be absorbed in the first-stage reheater = 890 B.t.u. The surface requirements may now be computed and are as follows:

Steam velocity, ft. per sec.	50	100	200
Surface per kw. in. First-stage reheater, sq. ft.	$\dfrac{890}{217 \times 7} = 0.59$	$\dfrac{890}{255 \times 7} = 0.50$	$\dfrac{890}{275 \times 7} = 0.46$

59 It is obvious that the steam velocity within the reheater has an appreciable affect on the surface requirements. Of course the coefficient 7 used on the gas side may be increased by increasing the gas velocity, but this involves an increase in the draft loss which may become prohibitive, whereas relatively a considerably larger steam-pressure drop in the reheaters will not be serious.

60 For the second-stage reheaters, the mean temperature difference between gas and steam is about one third that in the first reheater and therefore the same heat absorption, other conditions remaining the same, will require three to four times the surface, resulting in total reheating-surface requirements equal to or greater than that placed in the boiler, economizer, and superheater.

61 It appears, therefore, that two-stage reheating may require an excessive amount of surface unless it is found feasible to absorb from one-third to one-half of the energy for reheating directly from furnace radiation. Conservative design would thus seem to dictate the adoption of the combination cycle having one reheating stage and probably two or three bleeding stages.

DISCUSSION [1]

F. H. Rosencrants.[2] The author suggests under the heading Parallel Feed Heating that the highest efficiency might be obtained by splitting the feedwater into two parts, utilizing one in the economizer and one in the bleeder heater system. He points out that to work efficiently the economizer must receive reasonably cold water. The point seems to have been overlooked that this is also a requirement for the highest efficiency of feedwater heating by extracted steam. It has been well established that the turbine room can make better use of the low range of heating of feedwater by means of extracted steam than can the boiler room by means of flue gas. As a result of this the usual practice is to heat the feedwater to from 180 deg. to 225 deg. fahr. with bled steam before passing it to the economizer. Recent calculations have shown rather conclusively that the highest efficiency was obtainable by heating both air and feedwater with steam extracted at a low pressure, completing the heating of the feedwater with flue gas in an economizer. The cost of the air preheaters did not, however, make this commercially attractive.

W. J. Wohlenberg.[3] The authors have treated their problem in a most comprehensive manner, particularly for all possible ideal cycles to which practice may look as a standard. If machines can be built with uniformly high internal machine efficiencies throughout the range of conditions, there would be little left for argument. Since we have not perfect machines, certain questions arise.

The first of these is in connection with Fig. 20, in which curve D for the regenerative cycle falls considerably below curve A for the Rankine cycle. In the present state of development there may be certain losses in connection with stage heaters which will tend to decrease the machine efficiency of the regenerative turbine. On the other hand, the reduction of steam loading in the last rows of blades should materially increase the efficiency of that section, and it seems, therefore, that the internal efficiency of the regenerative machine should be nearly equal to or greater than that of the Rankine-cycle machine.

Undoubtedly the lower turbo-generator efficiency assigned to the regenerative machine is due to basing the available energy for the cycle on an infinite rather than on a definite number of bleeding stages. That is, efficiencies taken from curve D correct

[1] Complete discussion of four papers preceding is followed, on pages 802–823, by authors' closures.

[2] Discussion refers to Paper No. 1913 *a*, p. 644.

[3] Discussion refers to Paper No. 1913 *b*, p. 663.

for imperfections in the machine as well as for imperfections in the actual cycle as compared to the ideal. The term as used is thus a combination of cycle and machine-loss correction coefficient, instead of simply a machine efficiency. The same arguments probably apply to consideration of curves B and E.

The next question deals with the extrapolation of present test data on internal machine efficiencies to determine expected values beyond the range of present practice, particularly when reheating is to be used. This involves consideration of the conditions influencing internal turbine losses, which are largely determined by the mean condition of the steam as outlined in Appendix No. 4 of the writer's paper.[1] With proper precautions, the method there outlined should lead to accurate determinations of internal machine efficiencies.

The application of this method derives the curves in Fig. 18[2] for the various cycles indicated. It will be noted that all curves appear in their proper relation, and the variation of trends appears as it logically should. The Rankine- and regenerative-cycle curves are steeper than the others, and the slope decreases with increase of reheating in the various cycles. All curves shown, excepting that for the high-temperature cycles, have steeper slopes than the corresponding curves of Fig. 20, and the divergence of slope between the regenerative and the various reheating cycles is considerably greater in Fig. 18[2] than is the case with the author's curves. This difference alone will account in large measure for the more advantageous position that reheating assumes in the writer's paper above mentioned. Which set of curves is more nearly correct probably cannot be determined until more data under high-pressure conditions are available.

Referring to Par. 59, reheaters are assumed to absorb 3700 B.t.u. per sq. ft. per hr. A recent analysis indicates this value to be too high if both superheater and reheater are placed in the convection zone and screened from direct furnace radiation. For such conditions 2000 B.t.u. seems a more suitable figure for the reheater. In a radiant-type reheater the absorption would be at a much higher rate.

In Par. 92 are recommendations for the extraction of moisture from the turbine. Undoubtedly if a practical device were found which would continuously maintain the steam in the turbine in a dry gaseous state, reheating would be much less important. However, beyond the point of saturation in the expansion, moisture will appear in each row of blades even if the steam entering the row is dry. To approximate the gaseous condition through-

[1] Reheating in Central Stations. See page 741.

[2] The Fig. 18 referred to here is the Fig. 18 used in the closure of the discussion of the writer's paper on Reheating in Central Stations. See page 820.

out it would be necessary to install a large number of moisture-extraction stages, which seems impracticable. Reheating obviously moves the saturation point much further along the expansion line. It is logical to assume that no practicable moisture-extraction device could improve the internal machine efficiency to the same degree as appears possible by reheating. Here again the importance of the influence depends upon which of the curves under discussion is more nearly correct.

In this connection, it is highly important to have accurate data on the influence of conditions on internal turbine losses. It is apparent that, for the conditions chosen, the results shown in the first parts of the papers, both the authors' and of writer's,[1] are not debatable. However, in relating ideal and actual performance, debatable differences appear in the trends for machine efficiencies adopted. When considering high pressures, the field of ignorance is still so wide that various possible, and apparently logical, assumptions dictate serious differences in design. It appears, therefore, that a careful research in this field is at this time of great importance to the central station. Furthermore, no clear-cut direction may be given in new designs until these data are available. Whether such data may be made available other than by cut-and-try methods is an open question.

F. H. Rosencrants.[2] The study made by the authors eliminates, from the commercial standpoint, all cycles except the regenerative cycle D and the regenerative-reheat cycle E. Of these the regenerative cycle results in a simple and cheaper plant. The regenerative-reheat cycle must then justify itself by a substantial fuel saving to render it commercially superior.

The authors have limited the regenerative feature in both cycles D and E to that part of the cycle lying below the intersection of the adiabatic-expansion line and the saturation line. The upward slope of the efficiency curves for cycle D where they terminate at the saturation line in Fig. 11 shows that cycle D may be considerably improved by extending the regenerative process into the superheat region. The efficiency of this cycle will be a maximum when the regenerative process is extended into the superheat region to the limit of feedwater heating imposed by boiler-pressure saturation temperature.

Increasing the initial steam temperature from 700 to 800 deg. fahr. affects the efficiency of regenerative cycle D slightly, as stated by the authors, only for the calculations made in the paper. Had the regenerative process been carried to the limits noted above, the efficiency curves for the two temperatures

[1] Reheating in Central Stations. See page 741.
[2] Discussion refers to Paper No. 1913 *b*, p. 663.

shown in Figs. 10 and 12 for cycle D would have been much further apart.

In Part II of the paper, in analysis of plants operating on cycles D and E, the authors employ six more or less independent variables and processes as contributing to the economies, indicated as follows:

a Steam pressure
b Steam temperature
c Reheating steam after a partial expansion
d Heating feedwater with bled steam
e Heating feedwater with flue gas
f Heating combustion air with flue gas.

They might also have added the heating of combustion air with steam bled from the main turbine.

This process, as well as processes *d*, *e*, and *f*, aims to reduce the heat losses to condenser and stack by intercepting heat on its way to these points and returning it to the system via the feedwater and combustion air. The amount of heat that can be thus saved is limited by the heat-absorbing capacity of the air and water. The boiler room is deprived of this heat-absorbing capacity to the extent of its utilization in the turbine room. Since for each degree of temperature rise the water will absorb a little over twice as much as the air, and the capacity of the air is a little over 90 per cent of the heat content of the flue gas per degree, the combustion air has ample capacity to absorb as much heat from the flue gas as is commercially economical. The turbine room is thus free to utilize to the fullest extent practicable the heat-absorbing capacity of the feedwater. The ability to use highly preheated air in the combustion process is presupposed, as is the development of air heaters to accomplish the desired results on a commercially economic scale. The writer has considered the heating of combustion air with bled steam and the heating of feedwater with bled steam and flue gas in his discussion of Mr. Robinson's paper on the Margin of Improvement in Central Station Practice.[1]

The energy generated per unit volume of steam at the exhaust, mentioned in Par. 35, will be a maximum for its regenerative cycle, and will increase with the extension of the regenerative process up to the limit of boiler feed heating with bled steam, i.e., up to boiler-pressure saturation temperature. While the cost of the heaters for the regenerative process is justified in part in Par. 96 by the fact that the surface in the main condenser may be correspondingly reduced, dimensions of turbines at present are limited at the low-pressure end by mechanical difficulties rather than by economic considerations. Hence re-

[1] See page 644.

lieving the low-pressure end of part of the normal weight of steam handled may possibly enable it economically to use a higher vacuum, and therefore justify the cost of the condensing equipment to produce it. In such a case the condenser with bleed heaters may be as large as the condenser without them.

The authors state that to date only the surface type of heater has been used. The writer's company has adopted a contact type of heater in four plants now being designed, for absorbing steam from the middle one of three bleed points. These heaters discharge to the hot-water surge tank. They are much cheaper than the surface-type heater, and form an excellent point for deaëration of the feedwater. They are also a convenient point for discharge of condensate from such equipment as high-heat-level condensers of the evaporator plant, the condensate from the high-pressure bleeder heaters, and from various drips and drains.

While the labor of computing the economical heating surface and corresponding boiler-plant efficiency for each cycle would be arduous, as pointed out in Par. 40, the assumption of the same boiler-room efficiency for all cases detracts from the value of the paper. It is certain that it will cost more to obtain a given boiler-room efficiency for cycle D, which returns the feedwater to the boiler at the maximum temperature of all the cycles, than to obtain the same efficiency when using the Rankine cycle and returning the water at 75 deg. fahr. Fortunately the temperatures of the water returning to the boiler room in cycles D and E are not very different. The difference is in favor of cycle E, so far as boiler-plant efficiency is concerned.

Par. 90 suggests the use of a reheater, using steam whose heat at the throttle is 100 deg. in excess of that desired, to reheat the steam after partial expansion in the turbine. It may be asked if the turbine is the limiting factor so far as temperature is concerned. It seems to be a fact that the turbine can utilize steam at as high a temperature as present-day piping and superheaters can safely supply. This being true, any transfer of heat from steam at throttle pressure to steam at lower pressure can obviously only result in a loss of economy.

J. N. Landis.[1] Fig. 4 as drawn by the authors, while satisfactory for computing the cycle efficiency, is not a temperature-entropy diagram for the regenerative cycle. This fact is evident when one considers what the fundamental agreement is for which the ordinary temperature-entropy diagram is drawn, i.e., that all parts of the working fluid traverse identically the same cycle. With the regenerative cycle, the working fluid may

[1] Brooklyn Edison Co., Brooklyn, N. Y. Discussion refers to Paper No. 1913*b*, p. 663.

be divided into several parts, one of which traverses the ordinary Rankine cycle, while the others are bled out of the turbine at pressures above that in the condenser.

Referring to Fig. 4, the line *ab* represents addition of heat at constant pressure, the line from *b* to the saturation line represents an isothermal expansion, the line from the saturation line to the point *c* is a line of superheat at constant pressure, *cd* a line of adiabatic expansion, and *ea* an isothermal contraction. It is seen that all of the lines with the exception of *de* have a physical interpretation. To represent each change of state on the temperature-entropy diagram a line is necessary; therefore, to represent the changes of state of particles of steam passing from different initial conditions through an infinite number of heaters would require an infinite number of lines, all of the changes being different. The single line *de* cannot in itself represent the required infinite number of lines, therefore it can have no physical interpretation. Hence Fig. 4 is not a proper temperature-entropy representation for the regenerative cycle, for with the proper representation all lines can be interpreted physically.

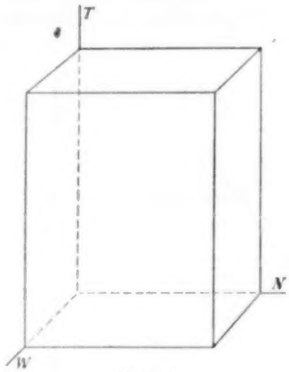

FIG. 2

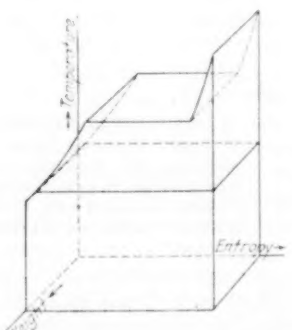

FIG. 1 THREE-DIMENSIONAL TEMPERATURE-ENTROPY DIAGRAM FOR THE ORDINARY RANKINE CYCLE

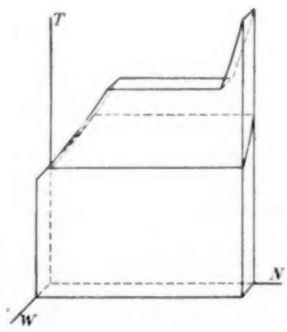

FIG. 3 TEMPERATURE-ENTROPY DIAGRAM OF THAT PORTION OF STEAM WHICH IS BLED OUT OF THE TURBINE IN THE REGENERATIVE CYCLE

It is suggested for the purpose of representing any cycle wherein different parts of the working fluid traverse different

cycles that the temperature-entropy diagram be drawn in three dimensions rather than in two as heretofore. Fig. 1 is the three-dimensional temperature-entropy diagram for the ordinary Rankine cycle. It will be evident from the following consideration how this can be drawn. Conceive of Fig. 2 as representing sheets of coördinate paper so stacked in the corner of a box as to stand parallel to the plane *TN*. Now let one pound of the working fluid be divided into the same number of parts as there are sheets of paper. If on each sheet a temperature-entropy diagram be drawn for one of the great number of parts into which the fluid has been divided, a great number of exactly identical diagrams will result. These when stacked together

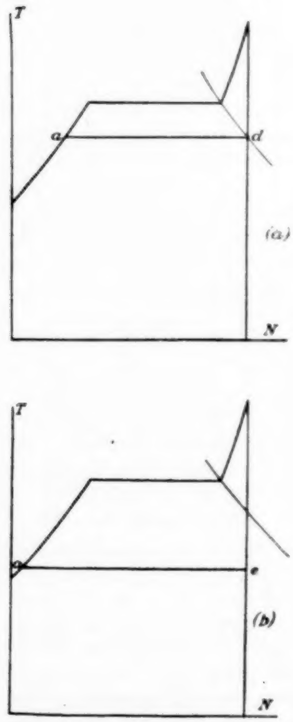

FIG. 4

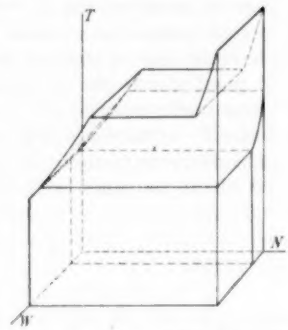

FIG. 5 SINGLE TEMPERATURE-ENTROPY DIAGRAM REPRESENTING THE CHANGES IN THE ENTIRE STEAM WHICH PASSES THROUGH THE TURBINE IN THE REGENERATIVE CYCLE

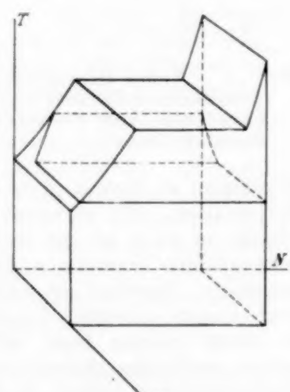

FIG. 6 VIEW OF FIG. 5 FROM A DIFFERENT ANGLE

again in the corner of the box give Fig. 1, which has depth to represent the fact that a specific weight of working fluid is under consideration. It is only because in the ordinary Rankine cycle all parts of the working fluid traverse the identical cycle that it is possible to ignore the element of depth and to draw the temperature-entropy diagram in but two dimensions. When using three dimensions to represent a temperature-entropy diagram, the units along the horizontal axis in the plane of the paper will represent units of entropy measured in B.t.u. per deg. fahr. per lb. Units along the vertical axis will represent deg. fahr. absolute, and the units along the horizontal axis, perpendicular to the plane of the paper, will represent pounds. The product of these three dimensions will give B.t.u.

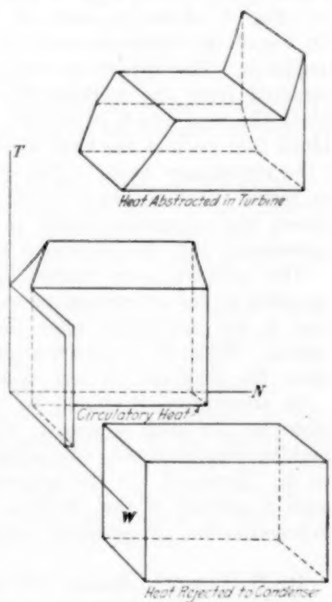

Fig. 7 Component Parts of Fig. 6 After Having Been Separated and Displaced

Fig. 3 shows the temperature-entropy diagram of that portion of steam which is bled out of the turbine in the regenerative cycle. This figure is based on the following reasoning: With Fig. 2 again in mind, conceive of the fluid as divided into a number of parts equal to the number of sheets of coordinate paper. Upon the sheet immediately adjacent to the plane TN the temperature-entropy diagram for the minute portion of steam bled out at the first or highest pressure heater is drawn. This particular diagram is shown in Fig. 4 (*a*). Upon the second sheet imagine the temperature-entropy diagram drawn for the steam bled out at the second heater. This diagram will differ from that in Fig. 4 (*a*) only in that the isothermal contraction line will be at a slightly lower temperature than the line from *d* to *a* in Fig. 4 (*a*). On each sheet, in order, a temperature-entropy diagram for the small portion of steam bled out at each sucessive heater is drawn. The diagram for the small portion of steam bled out at the last heater is shown in Fig. 4 (*b*) and is practically identical with the diagram for a minute portion passing through the condenser. Figs. 4 (*a*) and 4 (*b*) constitute limits, in that all of the isothermal contraction lines on the many diagrams fall between the line *da* in Fig.

4 (*a*) and *ea* in Fig. 4 (*b*). If now all of the diagrams be again stacked, in order, Fig. 3 results.

The diagram for that portion of steam which passes through the condenser in the regenerative cycle is of the same type as the diagram in Fig. 1, excepting that it has a lesser depth by the amount of the depth assigned to Fig. 3. If a single temperature-entropy diagram is desired which will represent the changes in the entire steam which passes through the turbine in the regenerative cycle, Figs. 1 and 3 can be combined as in Fig. 5. Fig. 6 shows a view of Fig. 5 from a different angle. In Fig. 7 the different parts of Fig. 6 have been separated and displaced. The uppermost block represents the heat which is abstracted from the working fluid in the turbine, the lowermost block the heat which is rejected to the condenser, and the central block that part of the heat which Mr. Orrok has so aptly termed "the circulatory heat." The efficiency of the regenerative cycle in terms of these blocks of heat is expressed by the ratio between the uppermost block as numerator and the sum of the uppermost and the lowermost blocks as denominator.

The authors have limited themselves to a consideration of bleeding at the saturation point, and state that their representation is not applicable when there is bleeding in the superheat region. With the representation here proposed, it is possible to draw the diagram for any cycle and conditions whatever.

In conçlusion, it should be understood that when it is possible to use such a simple device as the authors' Fig. 4 in making computations, advantage should most certainly be taken of it. However, for the purpose of analyzing various cycles to gain a picture of the physical changes undergone, a system of representation of perfectly general application should be used.

R. B. Purdy.[1] Except for steam conditions, the Marysville and Connors Creek plants of The Detroit Edison Company are of essentially identical design. Because of this they offer an excellent opportunity to demonstrate how closely the theoretical gains in economy due to higher steam temperatures and pressures are approached in actual practice.

Table 1 shows the operating conditions at the two plants. Table 2 sets forth the economies actually obtained, and the same economies after corrections have been applied. These tabulations are weighted averages of figures taken from monthly reports for a period of twelve months.

The actual thermal improvement shown by Marysville is 4 per cent. This improvement, however, does not represent the improvement due to high steam temperature and pressure, as it

[1] The Detroit Edison Co., Detroit, Mich. Discussion refers to Paper No. 1913 *b*, p. 663.

will be noticed that the back pressure, boiler efficiency, turbine efficiency ratio, and auxiliary use are not the same at both plants. If the exhaust pressure at Marysville were the same as at Connors Creek, the performance would be somewhat worse.

TABLE 1 OPERATING FIGURES OF MARYSVILLE AND CONNORS CREEK PLANTS

	Connors Creek	Marysville
Steam-pressure, lb. per sq. in. gage	220.	284.
Exhaust pressure, in. Hg abs	1.218	1.109
Steam temperature, deg. fahr.	598.	658.
Average monthly load factor	0.546	0.571

Applying average corrections for variation of turbine steam rate with exhaust pressure gives the Marysville results in line 4, Table 2.

A second correction is made because of the better Rankine-cycle efficiency ratio of the Marysville turbines. If the Marysville machines were reduced to the same efficiency ratio as the Connors Creek machines the plant economies would be as shown in line 5, Table 2.

A third correction is applied to reduce the Marysville boiler-room efficiency to that of the Connors Creek. Although this increases the thermal cost of power at Marysville it does not change the steam rate. Line 6, Table 2, gives these results.

A fourth correction is for auxiliary use. The value of this quantity for Marysville has been established on the assumption that the auxiliary use per kilowatt-hour of output is the same as at Connors Creek except for boiler-feed-pumping energy.

The auxiliary use for the two plants, in pounds of steam per kilowatt-hour of net plant output, is as follows:

	Total	Boiler Feed	All Others
Connors Creek	1.33	0.33	1.00
Marysville	1.64	0.49	1.15

The auxiliary use, aside from boiler-feed pumping, is higher at Marysville than at Connors Creek, because the Marysville plant is of relatively small capacity, being only the first section of a plant which will ultimately be much larger. Connors Creek is a complete unit. Boiler-feed pumping is, of course, different for the two plants by reason of the higher steam pressure at Marysville. To reduce the Marysville auxiliary use to Connors Creek conditions, the Marysville boiler-feed pumping (0.49) has been added to all other auxiliary uses for Connors Creek (1.00), giving the total figure of 1.49 lb. per kw-hr., which is 0.15 lb. per kw-hr. less than the actual. Correction for this quantity affects the plant steam rate, but not the turbine steam

TABLE 2 PLANT ECONOMIES OF MARYSVILLE AND CONNORS CREEK PLANTS

	B.t.u. per kw-hr.	Plant steam rate, lb.	Turbine steam rate, lb.	Theoretical Rankine steam rate, at 1 in. Hg back pressure	Efficiency ratio	Boiler-room efficiency, per cent	Factor of evaporation	Auxiliary use, lb. per kw-hr.
1 Connors Creek (actual)	19,625	12.75	11.42	8.03	71.8	74.5	1.185	1.33
2 Marysville (actual)	18,861	12.11	10.47	7.55	72.95	75.3	1.210	1.64
3 Per cent improvement by Marysville	4.0	5.0	9.0	6.0	.....	....	.....	....
4 Marysville corrected to Connors Creek back pressure	19,015	12.21	10.57	....	.....	....	.....	....
5 Marysville corrected to Connors Creek back pressure and efficiency ratio	19,280	12.38	10.74	....	.....	....	.....	....
6 Marysville corrected to Connors Creek back pressure, efficiency ratio, and boiler-room efficiency	19,470	12.38	10.74	....	.....	....	.....	....
7 Marysville corrected to Connors Creek back pressure, efficiency ratio, boiler efficiency and auxiliary use (except boiler feed)	19,240	12.23	10.74	....	.....	....	.....	1.49
8 Improvement by Marysville	2.0	4.3	6.3	6.0	.....	....	.....	....

rate; this correction consists in substracting 0.15 from the plant steam rate as corrected for previous items. Results appear in line 7, Table 2.

The net result of these corrections, shown in line 7, Table 2, is to establish operating economies for Marysville which would be obtained if that plant were operating with its equipment and load conditions identical with those prevailing at Connors Creek. Comparing these economies with the actual Connors Creek economies the percentage of improvement, shown in line 8, Table 2, is that due only to higher steam temperature and pressure.

If all conditions at the two plants except steam temperature and pressure were the same, the maximum possible improvement would be shown by the ideal Rankine-cycle steam rate as 6 per cent. The improvement in plant steam rate (4.3 per cent) is below the maximum possible gain, largely because of the greater energy required for boiler-feed pumping. If the Marysville boiler-

feed work were the same as at Connors Creek, the improvement would be 5.3 per cent. The improvement in B.t.u. per kw-hr. (2.0 per cent) is below the improvement in plant steam rate (4.3 per cent) largely because of a higher factor of evaporation at Marysville. If the Marysville factor of evaporation were the same as at Connors Creek, the improvement would be 4.0 per cent.

It is of interest to note that the authors show the improvement in B.t.u. per kw-hr. from 220 lb. per sq. in. gage to 284 lb. per sq. in. gage to be 2.4 per cent. This compares well with the figure of 2.0 per cent obtained in the above comparison.

V. S. Alden.[1] The writer believes that Professors Hirshfeld and Ellenwood have not given the regenerative cycle full credit and that its efficiency as given in their paper has been shown too low in Fig. 20. If, in a particular turbine, the Rankine-cycle ratios for the different individual stages are taken, the efficiency of the first stage may be 77 per cent and that of the third or fourth stage as high as 84 per cent. After the fourth stage, the efficiency of each succeeding stage is lower, down to the last stage. Taking into account the leaving losses the efficiency of this stage may be as low as 67 per cent. It is quite evident that if instead of the efficiency ratio of the different individual stages we take the cumulative Rankine-cycle efficiency ratio, the steam that is bled out beyond the fourth stage, i.e., the stage of maximum efficiency, will operate with the highest Rankine-cycle efficiency ratio, and on a higher ratio than the steam that passes all the way through the turbine to the condenser.

The main losses at the head end of the turbine, such as leakage losses past the high-pressure packing, throttling losses and disk-friction losses, are fairly constant in value. Expressed on a percentage basis, they vary inversely as the volume of the steam flowing from the turbine. Thus the percentage of loss at the head end decreases as the percentage of steam bled from the turbine increases. The foregoing and other reasons lead the writer to doubt the rather low value of efficiency given for the regenerative cycle by Professors Hirshfeld and Ellenwood.

It is commonly held that the most effective use of the regenerative cycle and the bleeding of steam from the early stages of the turbine is in unloading the later stages of the turbine and decreasing the leaving losses. This idea is a fallacy. The proper procedure with a standard turbine from which a considerable amount of steam is bled out of the early stages is not to decrease the steam flowing through the last stages as compared with operation without bleeding, but to increase the quantity of steam passing through the early stages. Comparatively minor modifications

[1] Discussion refers to Paper No. 1913*b*, p. 663.

in the early stages of the machine will permit this and lead to a considerable increase in the maximum capacity of the turbine.

In regard to the use of high steam pressures and high temperatures the writer has compared designs for a new station based on 205 lb. pressure at the throttle and 700 deg. fahr. temperature, with designs for the same station based on 365 lb. and 700 deg. The high-pressure station had four-stage bleeding and the low-pressure station three-stage bleeding, feedwater being heated to 374 and 291 deg. fahr. in the respective cases. The high-pressure station with identical boiler equipment has seven per cent greater output than the low-pressure station, notwithstanding the fact that the volume of steam going through to the condenser is $2\frac{1}{2}$ per cent less. The cost per kilowatt of maximum installed station capacity for the high-pressure layout is at least two per cent less than for the low-pressure station, by reason of the greater capacity from both boiler-house and turbine units due to the higher pressure and four-stage bleeding.

Frank S. Clark.[1] When required to actually determine the cycle or combination of cycles to be adopted for a new station, the extent to which the advantages of these cycles are to be made available, and the possible future developments which it may be desired to incorporate later, the designer must take into account certain factors other than those considered by the authors, of which the following are some examples.

As the economic application of possible methods for reducing fuel consumption is governed largely by the available load factor, it appears practicable to employ greater refinement for part of the total station capacity than for the remainder of that capacity. Thus, with some load characteristics it may be advisable to employ 1000-lb. steam with the reheating-regenerative cycle for such part of the capacity as may be operated continuously at practically 100 per cent load factor, and to employ a lower steam pressure with only a regenerative cycle for the less constant portion of the load.

The authors limit the discussion of the regenerative cycle to the bleeding of saturated steam only. The extension of bleeding into the zone of superheat apparently would improve the overall efficiency still further and would also considerably affect conclusions as to the advantages of reheating in combination with the regenerative cycle.

The practicability of employing in turbines now available the proposed high pressures without reheating, is still open to question. There is a decided variation in assumptions as to the decrease in turbine efficiency with increasing pressure, and this has been responsible for a similar variation in prediction as to

[1] Discussion refers to Paper No. 1913*b*, p. 663.

the economic value of reheating the steam. Definite conclusions on this matter must evidently await actual determination of the efficiency and serviceability of turbines with extremely wet steam in the lower stages, or the demonstration of a turbine design which will mechanically eliminate the moisture as anticipated by the authors.

If we permit ourselves to consider undeveloped possibilities, it appears that there would be both theoretical and practical advantages in reheating at the turbine by means of high-pressure steam or some other medium such as mercury vapor, which would provide the required heat by condensing at constant temperature. The temperature of reheating would thus be automatically controlled, the points of reheating could be chosen so that they would be most favorable to regenerative extraction, and the method would permit the isothermal-regenerative cycle to be more closely approximated.

As steam pressures are increased and refinements are introduced the cost of the equipment does not always increase by small increments but rather by relatively large steps. It is accordingly obvious that the economic selection of the pressure and cycle of operation may be quite different from that which would be indicated by smooth curves covering the entire range of possibilities.

Walter M. Keenan.[1] The method of determining the steam conditions at the various bleed points by means of a Mollier diagram raises a question as to the information that should be furnished by the manufacturer. In most cases the turbine manufacturer can offer to supply bleed points at, say, five stages of the turbine and the designer can select two or three of these points to be used as the extraction points for feedwater heating. In certain cases, however, where neither the complete design of the turbine has been worked out nor the water rates determined, the manufacturer will want to know the bleed temperatures required and the quantities of steam to be extracted; and the station designer will want to know what the water rates of the machine will be and the steam characteristics at various points in the turbine that might be available for the extraction, before calculations can be started. The result apparently is a vicious circle. If, therefore, turbine manufacturers supplied as part of their proposals Mollier or similar charts showing the steam conditions throughout the machine, the confusion and dissatisfaction that now prevails in giving complete and proper information for plant design would be largely eliminated.

In connection with the authors' computation tabulation, Fig. 7, the writer presents a method which he has used for three or four

[1] Discussion refers to Paper No. 1913 *c*, p. 713.

years to work out heat-balance problems. It is presented more as a supplement to the computation tabulation than as an entirely different method to be followed. As applied to the problem, outlined in the paper, of a turbine having two extraction points, this scheme uses three simple equations of the first degree, the three unknown quantities being

X = total lb. of steam flowing to the throttle
Y = lb. of steam extracted for the low-pressure heater
Z = lb. of steam extracted for the high-pressure heater.

The first equation is based on the assumption that the steam flowing through the turbine divides itself into three parts. The first part flows all the way through to the condenser, and its water rate WR_X is assumed to be the same as that of the machine without bleed. The power produced by this part will then be

$$\text{Kw.} = \frac{\text{Lb. of steam to condenser}}{WR_X}$$

The second part flows to the low-pressure bleed point. The water rate of this part is found by the formula

$$\text{Water rate} = WR_Y = \frac{3416}{(H_X - H_Y) \times E_g \times E_m}$$

where H_X = heat content of steam at throttle, B.t.u. per lb.
H_Y = heat content of steam at low-pressure bleed point, B.t.u. per lb.
E_g = generator efficiency
E_m = mechanical efficiency of the turbine.

The equivalent power of this bled steam then would be

$$\text{Kw.}_Y = \frac{\text{Lb. of steam to low-pressure bleed}}{WR_Y}$$

In a similar manner the water rate of the third part of the steam — that flowing to the high-pressure bleed point — is determined. The equivalent power of this steam is

$$\text{Kw.}_Z = \frac{\text{Lb. of steam to high-pressure bleed}}{WR_Z}$$

Equation [**2**], following, is based on equating the heat entering and leaving the low-pressure heater. Equation [**3**] is based on equating the heat entering and leaving the high-pressure heater. The equations are

$$\frac{X - Y - Z}{WR_X} + \frac{Y}{WR_Y} + \frac{Z}{WR_Z} = \text{Total output, kw.} \quad . \; . \; . \; . \; . \quad [\mathbf{1}]$$

$$(X - Y - Z)(t_2 - t_1) = Y[H_Y - (t_y - 32)] + Y(t_y - t_2) + Z(t_z - t_2) \quad [2]$$

$$X(t_3 - t_2) = Z[H_Z - (t_z - 32)] \quad [3]$$

where t_1 = initial temperature to low-pressure heater, deg. fahr.
t_2 = initial temperature to high-pressure heater, deg. fahr.
t_y = saturation temperature of steam at low-pressure heater pressure, deg. fahr.
t_z = saturation temperature of steam at high-pressure heater pressure, deg. fahr.
t_3 = final feedwater temperature of high-pressure heater, deg. fahr.
other notation being given above.

Similar equations readily may be made for any number of heaters to be installed. With this simple method any conditions imposed by the actual plant layout, such as desired temperature differences between steam and water, means of handling the heater condensate, etc., may be included in the equations. The solution of the equations will give values for the unknown quantities that are simultaneously in accord, without resorting to cut-and-try methods to make them agree.

The writer has worked out the case presented by the authors for the most economical load, using the above equations, and finds that his results vary somewhat from those obtained by the authors. A check of the authors results shows a discrepancy of some 0.5 per cent between the heat supplied by the bled steam and the heat absorbed by the feedwater. Just such an error is avoided by the use of simultaneous equations. In the example below, the figures are taken from the tabulation computation, Fig. 7, the figures in parentheses being the item number in the tabulation where the value used can be found.

Example. — From Fig. 7 we have the following values:
Heat at throttle (6) = 1356.5 B.t.u. = H_X
Heat at low-pressure nozzle (19) = 1079.0 B.t.u. = H_Y
Generator efficiency (4) = 0.964
Heat at high-pressure nozzle (19) = 1141.0 B.t.u. = H_Z
Water rate of turbine without bleed (2) = 9.53 lb. per kw-hr. = WR_X
Output at most economical load (1) = 18,680 kw. = P
Initial temperature to low-pressure heater (27) = 79 deg. = t_1
Initial temperature to high-pressure heater (27) = 150 deg. = t_2
Saturation temperature of steam at low-pressure heater pressure (15) = 157 deg. = t_y
Saturation temperature of steam at high-pressure heater pressure (15) = 217 deg. = t_z
Final temperature of feedwater (16) = 210 deg. = t_3

Calculation of WR_Y
Heat drop = $h_2 = H_X - H_Y$ = 1356.5 − 1079.0 = 277.5 B.t.u.
Heat per lb. of steam to produce 1 kw. = $h_2 \times E_g$ = 277.5 × 0.964 = 267.5 B.t.u.
WR_Y = 3416/267.5 = 12.75 lb. per kw.

Calculation of W_{RZ}

Heat drop $= h_1 = H_X - H_Z = 1356.5 - 1141.0 = 215.5$ B.t.u.

Heat per lb. of steam to produce 1 kw. $= 215.5 \times 0.964 = 208.0$ B.t.u.

$W_{RZ} = 3416/208.0 = 16.4$ lb. per kw.

Equation of Load

Substituting the necessary values in Equation [**1**], we have

$$\frac{X - Y - Z}{W_{RX}} + \frac{Y}{W_{RY}} + \frac{Z}{W_{RZ}} = 18{,}680$$

whence

$$X - 0.253Y - 0.419Z = 178{,}000.$$

Equation of Low-Pressure Heater

Substitution of the values in Equation [**2**] gives

$$(X - Y - Z)(150 - 79) = Y[1079 - (157 - 32)] + Y(157 - 150) + Z(217 - 150)$$

whence

$$X - 14.55\ Y - 1.94\ Z = 0$$

Equation of High-Pressure Heater

Substituting in Equation [**3**], we have

$$X(210 - 150) = Z[1141 - (217 - 32)]$$

whence

$$X - 15.92Z = 0$$

We thus have three simultaneous equations containing values of X, Y, and Z. Solving these we find, $X = 185{,}723$ lb. of steam per hour, $Y = 11{,}209$ lb. per hour, $Z = 11{,}666$ lb. per hour.

The foregoing figures may be checked as follows:

Total steam to throttle.. $= X$	$= 185{,}723$ lb. per hr.	
Total bleed steam........ $= Y + Z$	$= 22{,}875$ lb. per hr.	
Total steam to condenser $= X - (Y + Z)$	$= 162{,}848$ lb. per hr.	
B.t.u. added to main condensate $= 162{,}848(210 - 79)$		$= + 21{,}333{,}088$
B.t.u. added to l.p. condensate $= 11{,}209(210 - 157)$		$= + 594{,}077$
B.t.u. added to h.p. condensate $= 11{,}666(210 - 217)$		$= - 81{,}662$
Total net B.t.u. added (a)........................		$= 21{,}845{,}503$
B.t.u. added by l.p. bleed $= 11{,}209 \times 954$............		$= 10{,}693{,}386$
B.t.u. added by h.p. bleed $= 11{,}666 \times 956$............		$= 11{,}152{,}696$
Total B.t.u. added (b)..............................		$= 21{,}846{,}082$

The figures (a) and (b) are in substantial agreement.

A. G. Christie.[1] The authors have presented the most comprehensive treatment that has been published to date on the calculation of bleeder turbines. The methods are logical and simple. There is no reason why any engineer familiar with steam tables cannot make his heat-balance calculations with little difficulty by following the method outlined.

The writer has for some time used the same general methods in solving heat-balance problems with his classes. The use of the

[1] Discussion refers to Paper No. 1913*c*, p. 713.

Willans line to calculate inlet pressures as shown on Fig. 10 gives results in substantial agreement with test data. The writer has for some time drawn all expansion lines on the Mollier diagram to study turbine performance. These lines, however, had been drawn by empirical methods until Mr. Brown showed the writer test data which indicated that the efficiency of the superheated section of the expansion line is 10 per cent better than that of the saturated section. These are new data and seem to be applicable to pure Rateau turbines as well as to Parsons units. Incidentally this helps to explain the slight gain in internal efficiency with decreasing loads. As the authors point out, the expansion line takes a more complex form when first-stage Curtis wheels are used.

The authors have not called particular attention to the fact that the total heat of the steam at a given bleeder is substantially constant at all loads up to the "most efficient" load. This can be seen on Fig. 1 in the paper. Such evidence as is available tends further to indicate that this total heat at the bleeder point does not appreciably vary with slight changes in vacuum.

The authors state in Par. 25 that five per cent pressure drop is allowed at the extraction points. Is this figure based on performance data? It seems rather high.

The authors submit an empirical equation for mechanical losses in Par. 14. This, while a small factor in the problem, varies somewhat with the type of turbine. For instance, the mechanical losses of a turbine with water glands should naturally be higher than of one without them. Also the heating losses in bearings and thrusts with two-cylinder units may be higher than in single-cylinder units. It is usual to consider radiation losses as mechanical losses, for the equivalent heat has gone outside of the casing in both cases. Then two-cylinder machines with a long inter-cylinder steam pipe may need a slightly higher allowance for radiation than a single-cylinder unit.

Mechanical losses consist of friction losses in bearings and thrust, power to drive oil pump, governor, and water glands if used, and, as previously noted, radiation losses. It should be noted that these losses refer to the turbine only, as the bearing losses of the generator are included in the generator efficiency. Furthermore, since the speed of the turbine is nearly constant at all loads, these losses, being dependent on speed, should also remain constant at all loads with the exception of radiation, which is a small item in well-covered turbines. It will therefore be apparent that mechanical losses cannot be assumed as a constant percentage of output at all loads, but will be an increasing percentage at the lighter loads. A method of allowing for these variable losses is to estimate the equivalent horsepower loss at best load and at light load, and add these varying losses to the brake-horsepower output at each load. This gives the correct internal power.

Incidentally a study of internal efficiencies will lead to a more

rational basis for estimating heat per pound of steam rejected to the condenser and to a better analysis of condenser performance.

When turbines with high-pressure glands are considered, a correction must be made for the steam lost through the gland. When the guarantees include this steam, it should be deducted from the total steam before calculations are made of heat converted into internal work per pound of steam flowing through the turbine. A correction may be made for the work generated by the gland steam in the first stage of the turbine, though this is usually a small item. The authors indicate in Par. 16 some of the difficulties that confront manufacturers in furnishing guarantees on turbines that may be bled to heat feedwater. It would seem best to have the manufacturers first furnish estimates only on the performance of the particular unit until the complete bleeding scheme is fixed. Then final guarantees should be submitted for a turbine particularly designed to fit the specified conditions of bleeding. This may require some changes from standard designs by the manufacturers, but the increased efficiency will more than offset increased costs due to such changes, particularly in large-base-load and high-load-factor turbines.

The paper mentions the use of formulas, which are not presented, in the solution of heat-balance problems. The writer often has found it convenient to use some simple formulas for this purpose. These are given as a matter of record. Take the system shown in Fig. 6 and let

w = total steam entering throttle, lb. per hr.
H_1 = total heat per lb. of steam at throttle, B.t.u.
a and b = lb. per hr. of steam bled at high- and low-pressure bleeders
H_a and H_b = total heat of steam at bleeding point, B.t.u. per lb.
m and n = radiation losses in B.t.u. per lb. at high- and low-pressure heaters (assumed usually 5 B.t.u. per lb.)
q_a and q_b = heats of liquid corresponding to pressures in high- and low-pressure heaters, B.t.u. per lb.
t_a and t_b = saturation temperatures corresponding to pressures in high- and low-pressure heaters
K = terminal difference in each heater
t_c = temperature of condensate entering low-pressure heater
x = make-up added to condenser hotwell expressed as a decimal
H_e = total heat per lb. after complete expansion to exhaust pressure.

The following equations may be written down:

$$a(H_a - m - q_a) = w(1 + x)\{(t_a - K) - (t_b - K)\}$$
$$= w(1 + x)(t_a - t_b) \quad \ldots \ldots \ldots \ldots [1]$$

$$b(H_b - n - q_b) = \{w(1 + x) - a - b\}(t_b - K - t_e) - a(q_a - q_b) \quad \text{[II]}$$

$$(\text{Internal Kw.}) \times 3415 = (H_1 - H_e)(w - a - b) + a(H_1 - H_a) + b(H_1 - H_b) \quad \text{[III]}$$

Solve [I] for a in terms of w; substitute in [II] and solve for b in terms of w; then substitute for a and b in [III] and solve for w the total steam entering the turbine for the given output corresponding to the assumed internal kilowatts. All heat values in the above solution are taken from the Mollier diagram and steam tables just as in the author's paper. Similar formulas can be readily developed for additional heaters or for modified systems including evaporators, etc.

Item 43 in Fig. 7 is open to some question. In the preceding calculations the authors have figured the reduced output in each case when bleeding, corresponding to a given flow of steam with no bleeding. Item 43 attempts to correct back to the increased flow of steam when the same load is carried when bleeding as when not bleeding. While this item gives figures which are approximately correct, its method is not correct for it assumes that the flow of steam is proportional to the load, while a glance at a Willans line will show that this is not so.

The authors have presented a correct viewpoint in Pars. 59 and 60 in regard to air preheaters. Engineers in this country are only starting to realize the improvements in economy that may be attained by the use of air preheaters in conjunction with present equipment.

Linn Helander.[1] Approximate bleeder-performance data, such as Messrs. Brown and Drewry derive from estimated steam-expansion lines plotted on the Mollier diagram, should not be used to determine the relative merits of feed-heating turbines or of other equipment affecting the heat balance of a power station. For such determinations, as well as when final heat-balance calculations are to be made, accurate data should be obtained from the turbine manufacturer. Otherwise the assumptions forming the basis of the approximations may lead to unwarranted conclusions.

The authors' application of their method presupposes that reaction blading operates at ten per cent higher efficiency, and a combination of impulse and reaction blading at five per cent higher efficiency in the superheated region than in the saturated region. There is no thermodynamic justification for this supposition, as with either reaction or impulse blading the efficiency in the saturated region measured on the basis of the adiabatic heat drop is a function of the extent of the steam's expansion below the saturation line.

[1] Discussion refers to Paper No. 1913 c, p. 713.

A discussion of blading is extraneous to the subject of stage feedwater heating, but the implication that combination impulse and reaction turbines are not as well adapted to efficient bleeding as are reaction turbines is erroneous, as also is the explicit statement that impulse blading is inherently inefficient. In practical design, impulse wheels in the high-pressure ends of combination impulse and reaction machines are showing much higher efficiencies than those indicated by the expansion line the authors give in Fig. 5. Where impulse blades are used in reaction machines they replace less efficient reaction blading at the high-pressure end.

The statement that steam-jet air ejectors are wasteful of heat is misleading. The important consideration is not that high-temperature steam is degraded in the process of doing work, as it inevitably must be, but to determine whether the amount of steam passing to the condenser is increased or decreased when air ejectors replace air pumps. The important avenue of energy dissipation in the generator room is the condenser, where every pound of steam condensed represents a loss of approximately 950 B.t.u. If a motor-driven air pump is used the steam chargeable against it is the water rate of the main turbine multiplied into the power in kilowatts consumed by the air-pump motor. If the air pump is replaced by an air ejector, the steam consumed by the ejector will reduce the amount of steam bled from the low-pressure bleeding point and the steam chargeable against the ejector is the increased flow to the condenser due to this reduction in bled steam. When the pressure in the lowest-pressure heater is about that of the atmosphere the air ejector should be charged with approximately only one-half of the steam consumed by it. When the analysis is properly made, air ejectors usually will be found more desirable than air pumps for small stations where the complications of multiple-stage heating are not justified. For large stations the economies of air ejectors and air pumps are not far different, and a careful analysis should be made to determine which serve best the needs of a particular station. Air ejectors are well adapted to modern heat-balance layouts.

A. G. Christie.[1] Engineers are now devoting considerable thought to high pressures and temperatures and to various cycles for improving the station heat balance. The author has made a distinct contribution to the literature of this subject. His paper, combined with those of Professors Hirshfeld and Ellenwood,[2] and of Mr. Robinson,[3] form a series that will repay serious study.

The author has approached his analysis of cycles from a different

[1] Discussion refers to Paper No. 1913 *d*, p. 741.

[2] High-Pressure, Reheating, and Regenerating for Steam Power Plants. See page 663.

[3] The Margins of Possible Improvement in the Central-Station Steam Plant. See page 644.

standpoint than that of Professors Hirshfeld and Ellenwood. The same cycle efficiencies may be found from the curves of both papers, provided the same assumptions of pressure and bleeding temperatures are chosen. When the commercial application of the cycles is made, the two papers are somewhat at variance. Professors Hirshfeld and Ellenwood base their results on a series of turbo-generator efficiency curves shown in their Fig. 20 [1] and on certain assumptions of plant performance, such as, for instance, a constant value of 84 per cent for boiler and economizer efficiencies at all loads.

The author, on the other hand, assumes varying boiler efficiencies at various pressures as stated in Table 3. His curve of internal machine efficiency given in Fig. 10 is a novel method of representing turbo-generator efficiency that has never before come to the writer's attention. This curve gives efficiencies quite at variance with those on Fig. 20 in Professors Hirshfeld and Ellenwood's paper, and accounts in a large measure for the variation in final results. Let us see whether such a curve as the author's Fig. 10 is justified in theory, and then whether its trend is correct. The author has indicated in Appendix No. 4 that these data apply only to large turbo-generators operating at 29 in.

It is a well-known fact that the sections of a turbine using wet steam are less efficient than the sections using superheat. Messrs. Brown and Drewry indicate in their paper [2] that there is a difference of 10 per cent in efficiency between the two portions of a Parsons turbine. A Mollier diagram will show that a greater and greater proportion of the expansion takes place in the super-heat field as the adiabatic end point on the 29-in. vacuum line increases in value for the assumed inlet condition of 750 deg. fahr. It is logical, therefore, from theoretical considerations, to expect the internal machine efficiency to increase with increasing values of the adiabatic end point.

The author's curve, Fig. 10, is based on data from only a few turbines and some of these were compound and three-cylinder units. The author points out that many of the data on which the internal turbine efficiency may be based are known only to turbine designers. It will be of assistance to engineers to know whether these designers can confirm the author's claim that the efficiencies can be represented by such a curve as shown.

The writer has made an attempt to check this curve by making certain assumptions with data at hand. The points taken from several makes of turbines are somewhat scattered. A mean curve drawn roughly through these points shows a somewhat steeper slope than in Fig. 10 for end points above 870 B.t.u. This is quite possible as the upper points D, E and F in Fig. 10 are for compound

[1] Page 684.

[2] Economy Characteristics of Stage Feedwater Heating by Extraction. See page 713.

units where the mechanical and radiation losses may be greater than in single-cycle units. If additional allowance is made for these losses, the curves in Fig. 10 would become steeper and would conform closely to the writer's data.

The final performance curves in Fig. 30 of the Hirshfeld and Ellenwood paper and in Fig. 14 of the author's paper are at some variance, due to factors already discussed. However, the author does not include auxiliaries and generator losses, and, furthermore, has assumed a decreasing boiler efficiency with increasing pressure. He finds about 1000 lb. as the most economical point, while the Hirshfeld and Ellenwood curves have not reached a minimum, with the exception of the Rankine cycle, at 1200 lb.

A significant point is brought out in Fig. 14 of the author's paper in regard to the effect of higher temperatures. Increasing temperature from 750 deg. fahr. to 1000 deg. fahr. results in a decrease of 500 B.t.u. per kw-hr. in heat consumption. Will it be more economical to increase temperature at moderate pressures than to increase pressures to 1200 lb. or more?

It has been assumed in the discussions of the various cycles that no steam is bled for heating feedwater at pressures above the saturation line on the expansion curve. This has led some engineers to believe that bleeding of superheated steam is wrong and should not be done in any case. This is a mistaken idea that should be corrected. While it is true that there is not as great a relative gain in efficiency in bleeding superheated steam as saturated steam — as can be shown on a temperature-entropy diagram, — still there is a significant gain in many cases by bleeding such steam. The writer has had a number of calculations made on this point and has always shown a decreased B.t.u. per kw-hr. by such bleeding at high pressures in the superheat section. This gain is more apparent when few bleeding points are used. In fact, this plan has been adopted in some of the new central stations where desuperheaters are provided to take care of superheat at the bleeder heaters.

Linn Helander.[1] The need of extrapolation in calculations to determine the economy of turbines using reheated steam is lessened if the expansion is divided into one part from the condition of the steam at the throttle to its condition at the reheating point, and into another part from the reheating point to the condenser. Calculations then will show an improvement in economy of about 5 per cent, after deductions for losses in the reheater, by reheating to 700 deg. fahr., with initial steam conditions of 550 lb. pressure and 725 deg. fahr. total temperature. The best reheat pressure then lies between 100 and 125 lb. gage, occurring after approxi-

[1] Discussion refers to papers Nos. 1913 *b*, page 663, and 1913 *d*, page 741.

mately one-third of the adiabatic heat drop. These results agree neither with those found by Professors Hirshfeld and Ellenwood nor with those found by Professor Wohlenberg, and the assumptions forming the basis of their extrapolations should be considered.

The regenerative and reheating cycles proposed by Professors Hirshfeld and Ellenwood end the feedwater heating at the saturation point of the expansion line. There is no thermodynamic reason for this, since an ideal cycle, implying infinite feedwater heating surface, should heat the feedwater to the temperature of saturated steam in the boilers. The proposed regenerative cycles do not appear to form a satisfactory basis for extrapolation, and their use for that purpose confuses us in our understanding of the term "efficiency." That the small improvement in efficiency resulting from increasing the throttle temperature to 800 deg. fahr. is due to the fall in bleeder temperature as the throttle temperature rises, is true only for the type of diagram used by the authors. It is not true for a diagram representative of the best available combination reheating-regenerative cycle.

The temperature-entropy diagram can be simplified by drawing the diagrams for the steam bled for feed heating separate from the diagram for the steam passing to the condenser. If this is done, the temperature to which the feedwater may be raised will be seen to be independent of reheating, although reheating may influence the desirable feedwater temperature. Also the cycle of the steam bled after the reheating point is one using exhaust steam against a high back pressure, and its best reheat point would be at a higher pressure than that for a full-expansion cycle. Unless the bled steam is taken from a separate turbine designed for reheating at the most efficient point, the best reheating pressure will be a compromise between that best for bleeding and that best for the full-expansion cycle, with some sacrifice in economy. The best reheat pressure for a bleeding and reheating cycle is higher than for a full-expansion cycle. Also the improvement in efficiency due to reheating is lessened by increasing the number of stages of feedwater heating.

Only part of the gain from reheating in a practical station is due to the thermodynamic factors affecting the best reheat point; the remainder may be due to factors which cause an appreciable lowering of the best reheat point. Professor Wohlenberg's Fig. 3 on page 743 shows the influence on the efficiency of the station of varying the number of stages of reheating, adding in each case a total of 300 B.t.u. in the reheater. Fig. 4 (p. 743) shows a better reheat point with two-stage reheating, adding 150 B.t.u. per stage, than with single-stage reheating, adding 300 B.t.u. If maximum efficiencies for various numbers of stages of reheating are plotted, a different curve is obtained. Since most of the benefit of reheating not accounted for in the theoretical study accrues in the first

stage of reheating, a practical plant would probably find two-stage reheating undesirable.

Professor Wohlenberg uses the same feedwater temperature for all initial steam pressures. This may have affected adversely the efficiencies of the bleeding cycles for high-pressure steam. The efficiency curves for the combination bleeding and reheating cycles tend to flatten more rapidly at the higher steam pressures than the others. This probably has been accentuated by using the same reheat pressure for this cycle as for the non-bleeding cycle as well as by using a constant feedwater temperature.

Professors Hirshfeld and Ellenwood apply corrections for reheating to the overall efficiencies of the turbine, whereas reheating affects only that portion of the turbine beyond the reheat point. Also, the validity of their method of extrapolating regenerative efficiencies is not obvious. When regeneration is used the efficiency of that part of the cycle pertaining to the steam passing through to the condenser should be the same as defined by their curves of Fig. 7 (p. 673) for the Rankine cycle. That the curve of efficiencies of the regenerative cycle is below that for the Rankine cycle must be due to assigning to the bled steam lower efficiencies measured relatively to its theoretical capacity for doing work than apply to the steam passing to the condenser. Bleeding efficiency, defined as the ratio of work done by the bled steam to the work that theoretically can be done by this steam heating the feedwater to the same temperature it is heated practically but using infinite heating surface, depends on the arrangement and number of feedwater heaters and on the efficiency of that part of the turbine in which the bled steam expands.

Assuming that the regenerative-efficiency curve will lie parallel to the Rankine-cycle efficiency line involves the assumption that the efficiency of bleeding is a maximum at 1000 lb. initial steam pressure, and practically constant between 800 and 1200 lb. per sq. in. steam pressure, as shown by the accompanying Fig. 8, which the writer has calculated from the data of Table 5 of Professors Hirshfeld and Ellenwood's paper (p. 690). Since the efficiency of bleeding tends to increase with an increase in the number of heaters and the number of heaters increases with the initial steam pressure, this type of bleeding-efficiency curve is not unreasonable. The presentation, however, would have been more satisfying if the bleeding efficiency had been determined by calculation, in which case it would have been unnecessary to limit the temperature of the feedwater to the saturated-steam temperature in the turbine.

Professors Hirshfeld and Ellenwood reduce the turbine efficiency one per cent for a combination regenerative-reheating cycle, due to the lowering of the feedwater temperature consequent on higher reheating in the turbine. This is an error since the proper feedwater temperature should be independent of any reheating in

the turbine due to frictional and leakage losses, and such reheating would not be a determining factor of the proper feedwater temperature in a practical plant.

Professor Wohlenberg extrapolates efficiencies by plotting them against what he terms the adiabatic end point.[1] This method may show consistent results when applied to straight condensing turbines, but the application of data so obtained to turbines operating on a reheating cycle is not justified. The overall efficiency of a turbine, usually expressed in terms of work done divided by the total adiabatic heat drop, includes a reheat factor. Frictional and leak-

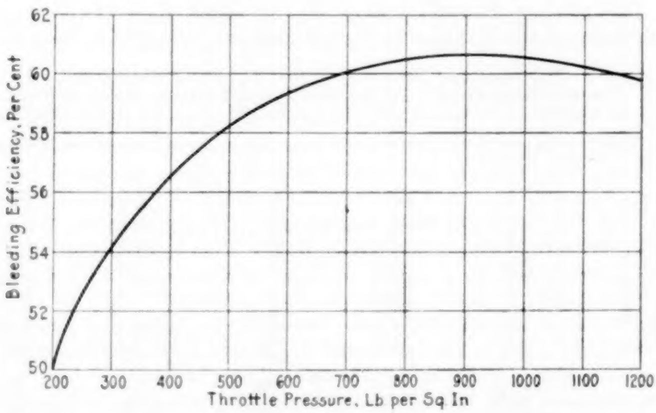

Fig. 8 Bleeding Efficiency at Various Throttle Pressures

(The bleeding efficiency plotted is defined as the ratio of the work done by the steam bled to the work it theoretically could do heating the feedwater to the same temperature it heats it practically but using infinite heating surface. Since this efficiency increases with the number of heaters it should increase with the initial steam pressure as does the above curve derived from Table 5 of Professors Hirshfeld and Ellenwood's paper.)

age losses in the turbine result in a degradation rather than a loss of energy, so that the steam expansion line follows one of increasing entropies rather than one of constant entropy. Efficiencies obtained from tests and based on the adiabatic heat drop to the exhaust pressure properly cannot be used when the entropy is shifted in calculating the heat available for conversion to work as is done at the point of reheating. It is well known that the efficiency of the low-pressure element of a compound turbine based on its apportionment of the adiabatic heat drop over the entire unit is greater than its efficiency as a separate turbine. If this higher efficiency is used to determine the work it does and the calculations are based on the state of the steam entering it, as when reheating, the results will credit the steam with doing more work than it

[1] See par. 15, page 749.

is capable of doing. The error is due to considering the heat equivalent of the internal losses as additional available energy, at the same time using an efficiency which includes the corrective factor for the reheating effect of these losses.

Professor Wohlenberg endeavors to avoid this error by charging the turbine both with the heat added in the reheater and with the heat equivalent to the friction and leakage losses in the turbine down to the pressure at which reheating occurs. The limiting case of this is when no heat is added in the reheater and all reheating of steam is due to internal losses in the turbine. The accuracy of his method applied to this limiting case may be tested.

The product $E_c \times E_i = E'_c \times E'_i$, in which E_c is the thermal efficiency of the Rankine cycle with adiabatic expansion from the

TABLE 3 COMPARISON OF EFFICIENCY FACTORS DETERMINED BY CONSIDERING THE CYCLE OF A STRAIGHT CONDENSING TURBINE AS A REHEAT CYCLE WITH EFFICIENCIES USED BY WOHLENBERG

(Initial steam conditions: Pressure, 600 lb. per sq. in.; total temperature, 700 deg. fahr.; vacuum, 29 in.)

Pressure at entrance of low-pressure element, lb. per sq. in.	Thermal efficiency of reheat cycle, per cent	Heat content of steam entering low-pressure element, B.t.u.	Adiabatic end point, B.t.u.	Efficiency corresponding to adiabatic end point from Wohlenberg's curve, per cent	Calculated efficiency factor based on reheat cycle, per cent
600	38.3	1349	851	76.2	76.2
100	38.0	1214	878	78.8	76.5
15	37.3	1110	910	80.6	78.0
5	36.6	1059	929	81.7	79.6
2.5	36.1	1030	941	82.1	80.5

initial state of the steam to the exhaust pressure, E_i is the overall internal efficiency of the turbine, and E'_c is the thermal efficiency of the reheat cycle which is made up of an adiabatic expansion to some pressure, a reheating at this pressure equivalent to the internal losses, and a subsequent adiabatic expansion to the exhaust pressure. E'_i is the efficiency factor which satisfies the above equation and should fall on the curve shown by Professor Wohlenberg's Fig. 10 (p. 747), assuming that his method is applicable to this limiting case. Table 3 of the writer's discussion evinces the divergence of efficiencies calculated for this limiting case and those given by Professor Wohlenberg.

A second test of accuracy is to segregate the efficiency of the high-pressure and low-pressure elements of a compound turbine and then calculate the efficiency of a combined unit reheating between elements to the initial heat content, in one case using Professor Wohlenberg's method and in the other determining the work done by each element separately. If the initial steam conditions are taken as 600 lb. per sq. in., 700 deg. total temperature, and the

reheat pressure as 180 lb. abs., the curve of efficiency given by Professor Wohlenberg should hold fairly accurately for the low-pressure element. The curve also should be applicable when the two elements operate as a single unit. If the data of the curve are sufficiently consistent, the work done by the low-pressure element may be determined and subtracted from the work done by the entire unit operating on a non-reheating cycle and the efficiency of the high-pressure element may be established. Calculations made in this way indicate that for the initial steam conditions assumed and 29 in. of vacuum, the efficiency of the high-pressure element would be about 50 per cent and that of the low-pressure element when no reheating is done would be 79.5 per cent. The high-pressure element of a commercial turbine would not have an efficiency as low as 50 per cent, and in the following analysis this efficiency is used for the high-pressure element only for the sake of consistency. Reheating obviously will not affect the efficiency of the high-pressure element. If the steam is reheated to its initial heat content the efficiency of the low-pressure element will be 81.1 per cent as determined from Professor Wohlenberg's curve and the work done by each pound of steam passing through it will be 348 B.t.u. The work done by the entire unit, therefore, will be 348 B.t.u. plus the work done by the high-pressure element, which is 63 B.t.u., making a total of 411 B.t.u. per pound of steam. This corresponds to a thermal efficiency of 30.1 per cent. Professor Wohlenberg's method would give a thermal efficiency of $0.811 \times 554 \div 1427 = 31.5$ per cent. There is an apparent discrepancy in the results. An aggregation of errors undoubtedly follows the indiscriminate use of efficiencies determined from the adiabatic end point for various types of cycles, including bleeding and multiple-stage reheating.

Like Professors Hirshfeld and Ellenwood, Professor Wohlenberg determines the best pressure for reheating from theoretical considerations. He ignores the fact that reheating is essential to lessen the moisture in the low-pressure stages. This error cannot be corrected by overall corrections of turbine efficiencies. Boiler-room efficiency need not drop at high pressures if air preheaters are used, either with or without economizers. Figs. 13 and 14 (pp. 750 and 751) of Professor Wohlenberg's paper indicate that powdered coal can be used more advantageously than stokers at high pressures. This is true, as he points out, only because he has penalized the stokers by reducing their efficiency at high pressures, while maintaining that of powdered coal by means of air preheaters. Stokers are well fitted to use preheated air and should not be penalized in this way.

B. N. Broido.[1] Fig. 8 (p. 674) of the paper by Professors Hirshfeld and Ellenwood, showing the efficiency of reheating, shows that

[1] Discussion refers to papers Nos. 1913 *b*, p. 663, and 1913 *d*, p. 741.

the maximum efficiency from reheating can be derived when the reheating occurs at a pressure of from 0.23 to 0.24 of the initial pressure — the higher figure for 1200 lb. pressure, the lower for 200 lb. pressure. This is true only for the conditions stated, and where the reheating is carried to the initial temperature of the live steam. In general, however, the benefit obtained from heat added to steam is usually greater the higher the steam pressure at which it is added. On the other hand, if it is desired to reheat the steam to the initial temperature, which is the most advantageous from a practical standpoint, the lower the steam pressure the greater is the heat that can be added. The best compromise between these two extremes gives the pressure with the maximum benefit of reheating, and it depends entirely upon the conditions. In certain cases it may even be advisable to bleed steam which is still superheated, and thus increase the total efficiency.

Par. 32 and Fig. 19 (pp. 677 and 682, respectively) lead to the conclusion that at the higher pressure the regenerative-reheating cycle is less efficient than the regenerative cycle alone; that is, reheating decreases rather than increases the efficiency. This does not do justice to reheating. The curve is based upon the ideal regenerative cycle, which involves bleeding the turbine at an infinite number of stages, and which can never be realized in practice. At present it is considered the most advantageous to bleed the turbine in two, or a maximum of three, stages. The first bleeding can occur at the point of expansion where the live steam has just lost its superheat. The remainder of the steam can then be reheated to a temperature sufficient to have the steam saturated just at the second bleeding, after which the steam can be reheated again. In this case we will get the maximum benefit that can be obtained in practice from the bleeding of a turbine, or the incomplete regenerative cycle. Considerable advantage will also be derived from the reheating, giving a total efficiency higher than could be obtained with the regenerative cycle alone.

In Fig. 30 (p. 703), at pressures of about 400 lb. the reheating cycle gives an increase of 5 per cent. This increase can be obtained with a comparatively simple arrangement which involves no complication and is easy to operate. The authors have brought out the difficulty of reheating the steam by the boiler gases, and the complication involved in returning the steam to the boiler and thence again to the turbine. A simpler arrangement would be to reheat the steam in a separately fired superheater.

It is generally believed that separately fired superheaters are inefficient. However, separately fired superheaters now are on the market which have water tubes in the furnace ahead of the superheater. These can be operated with high furnace temperatures and are very efficient, particularly so when the steam to be heated is of comparatively low temperature and, correspondingly, the exit gas temperature is also low. This would work out very satis-

factorily in a plant where the auxiliaries are driven electrically from the main switchboard, and where there is no exhaust steam available for reheating the water. The feedwater would then be preheated in the water surface of the separately fired superheater. This arrangement would give a high overall efficiency. It would not be complicated, as the separately fired superheater could be located in the turbine room and fired with oil or gas, or with coal. In the latter case it would be located in the turbine room between

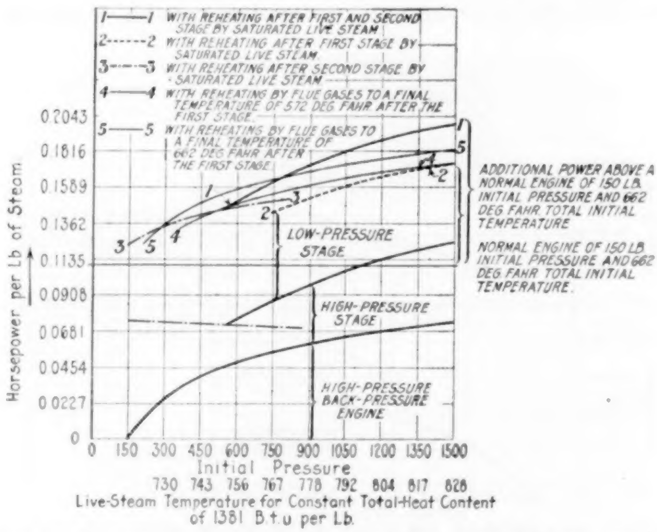

Fig. 9 Horsepower per Pound of Steam of a Plant Consisting of an Engine Built for 150 lb. Initial Pressure, Utilizing the Exhaust Steam of a High-Pressure Back-Pressure Engine. Also the Horsepower of Each Stage and the Influence of Different Methods of Reheating.

(Condenser vacuum in all cases 28 1/2 in.)

the boiler and turbine, with the furnace operated from the boiler room.

There are two methods of reheating steam which are not mentioned in either of the papers by Professors Hirshfeld and Ellenwood and by Professor Wohlenberg. The steam can be reheated by live steam from the main boiler. This is particularly advantageous for the highest pressures. Naturally, the steam cannot be reheated to the initial temperature of the live steam, but this method has many other advantages which makes it more desirable than others. It is not necessary to return the steam to the boiler room, as the reheater can be located near the turbine. The apparatus for reheating requires a comparatively small amount of heating sur-

face, as the reheating absorbed is the latent heat from the steam. This heat is easily given up, and the heat-transmission coefficient is high.

The most important advantage is the possibility of regulation. While some superheater designs now on the market give fairly constant temperatures at different ratings, the superheat is liable to vary due to fluctuation in the furnace. The temperature drop of the steam in the turbine will, of course, vary, depending upon the load, etc. If the steam is then sent back to a superheater installed in the boiler setting we have a third variation, so that the

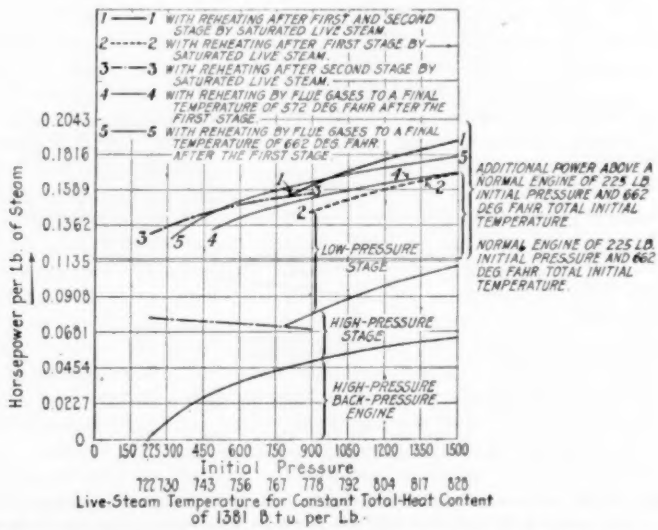

Fig. 10 Horsepower per Pound of Steam of a Plant Consisting of an Engine Built for 225 lb. Initial Pressure, Utilizing the Exhaust Steam of a High-Pressure Back-Pressure Engine. Also the Horsepower of Each Stage and the Influence of Different Methods of Reheating.

(Condenser vacuum in all cases 28 1/2 in.)

total fluctuation of the temperature of the reheated steam may become very serious. If the reheating is accomplished with live steam the regulation is much facilitated, as all that is necessary is to regulate one valve admitting the live steam to the reheater. This is a very important factor in favor of reheating by live steam.

For the same reason, objection should be raised to Professor Wohlenberg's suggestion to arrange the reheater in the furnace as a radiant-heat superheater. Heating surface placed directly in the furnace is subjected to all furnace fluctuations, and it is difficult,

if not impossible, to provide means for regulation. A radiant-type superheater is, therefore, in the opinion of the writer, entirely unsuitable for a reheater.

The second method for reheating steam is to install a separate high-pressure boiler in the boiler room, the steam from which would be used only for reheating. Except for the additional boiler, which must operate at a considerably higher pressure than the main boilers, this arrangement has all the advantages mentioned before.

Both papers in question are based on theoretical analysis, as, so far as is known to the writer, no plants in this country have

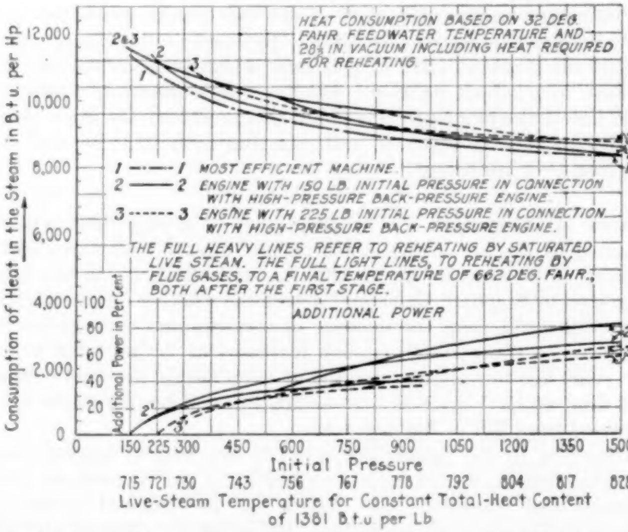

Fig. 11 Heat Consumption of a Most Efficient High-Pressure Installation; Heat Consumption of an Installation Consisting of an Engine for 150 and 225 lb. Initial Pressure in Connection with a High-Pressure Back-Pressure Engine; also Additional Power Obtainable as Compared with 150 lb. and 225 lb. Initial Pressure.

applied reheating. Considerable work in this connection has been done abroad, particularly by Dr. Schmidt and Mr. Hartmann. In 1910 a plant was built for a working pressure up to 900 lb. This plant was operated for a number of years and experiments made with reheating of the steam between the stages, or rather between the cylinders of the reciprocating engine used. The writer has obtained from Mr. Hartmann, who conducted most of the tests, curves showing the advantages of reheating both with live steam and with flue gases from the boiler.

Some of the curves already show the difference in efficiency that

may be expected between these two arrangements. Fig. 9 shows the horsepower per pound of steam in a plant consisting of an engine built for 150 lb. initial pressure, utilizing the exhaust steam of a high-pressure stage in accordance with the first-mentioned arrangement. It shows also the horsepower of each cylinder; the various curves are for different methods of reheating the steam as indicated.

Fig. 10 shows the horsepower per pound of steam of an engine built for 225 lb. pressure. Fig. 11 shows the heat consumption of a high-pressure plant for both reheating arrangements mentioned above. It also shows the additional power obtainable as compared with 150 lb. and 225 lb. initial pressure.

The writer understands that of the two plants which are now being built in this country for 1200 lb. pressure, one will be arranged so that the steam at 1200 lb. will enter a high-pressure turbine, the exhaust from which, at about 350 lb., will be discharged into the main line of the plant carrying about this pressure; the other plant will be an entirely separate unit, starting with 1200 lb. and discharging to a condenser.

John H. Lawrence.[1] The prophecies of savings that will be obtained by the use of high pressures and high temperatures are of interest, but before we go to 1200 lb., or to 3200 lb. as has been suggested in England, we should determine whether commensurate gains will result from the increase in pressure and temperature as compared to the gains from such increases in the past ten years. Ten years ago few plants operated at over 250 lb. pressure. Since that time we have gone as high as 400 lb., and temperatures have increased from 550 to 700 deg. fahr.

According to the figures presented in the various papers, considerable benefit should have resulted from the increase in temperature and pressure. The writer has investigated results of various plants operating under from 300 to 400 lb. pressure and 650 to 750 deg. temperature, and has compared them with stations operating at about 200 lb. pressure and about 100 deg. of superheat. There was no conclusive evidence that the increased pressure and temperature had increased the efficiency to any considerable extent. For instance, a certain plant operating at 250 lb. pressure and 600 deg. temperature with a not very favorable load factor has about the same B.t.u. per kw. as two or three other plants built at the same time and operating at from 300 to 350 lb. pressure and 550 to 750 deg. temperature. Another plant, operating at 215 lb. pressure and 125 deg. superheat or a total temperature of 500 deg., was only about 10 per cent higher in B.t.u. per kw. than some of the stations built in the last two or three years and which are rated as superpower stations. The only reason for the 10 per cent difference is the smaller size of the turbines as compared with the turbines of

[1] Discussion refers to papers Nos. 1913*b*, p. 663, and 1913*d*, p. 741.

the newer stations. Two of the turbines are 12,500-kw. and another is 20,000-kw. and their efficiency is less than 70 per cent. Furthermore, the auxiliaries for the largest part of the stations are steam-driven, non-condensing, with no stage bleeding. The boiler-room efficiency also is lower than would be expected today in a modern station. Notwithstanding these drawbacks, this station is almost as efficient as the later stations. If the turbine efficiency is corrected to the efficiency of the larger machines, and electrically driven auxiliaries are figured instead of steam-driven auxiliaries, the results are so close that a little carelessness in operating might easily send the high-pressure, high-temperature stations above the smaller station with low-pressure and low-temperature as regards the B.t.u. per kw. In Europe, one station operates at 475 lb. pressure with a temperature of 700 deg. It uses 20,000 B.t.u. per kw. A number of other large stations in Europe operate at over 300 lb. pressure, and their results are no more favorable than those in some of the smaller stations in the United States. The station operating under 475 lb. pressure has had its troubles, which have been the result of trying to obtain the last B.t.u. per kw. It has air heaters and has had trouble with the furnaces. There has also been turbine trouble due to the high temperature.

The writer's investigation of this subject is not complete, but all figures point to the fact that the gains due to high temperature and high pressure may not be commensurate with the trouble and expense involved. This may be due to operation or to some fault in design, but there is no evidence as yet that enormous gains will result from 600, 1200, and perhaps 3200 lb. pressure.

Geo. A. Orrok.[1] The several papers emphasize the advantages of circulatory heat, that is, heat taken from the exhaust side of the plant and turned back into the intake side. The more of this circulatory heat that can be used the greater will be the efficiency of the plant and the lower the loss of heat to the condenser.

It is worth noticing that although the several papers have been worked from somewhat different suppositions, the curves themselves show comparatively little difference. The difference between 700 lb. pressure and 1000 lb. as the best pressure for a station is small, considering the fact that the papers give only theoretical calculations based on quite different initial assumptions.

F. Hodgkinson.[2] Regeneration by bleeding the prime mover in stages for heating feedwater and by reheating partially expanded steam is old practice. The surprising thing is that engineers have only turned their attention to such processes, particularly regeneration in connection with steam turbines, during the last two years.

[1] Discussion refers to papers Nos. 1913 *b*, p. 663, and 1913 *d*, p. 741.

[2] Discussion refers to papers Nos. 1913 *b*, p. 663, 1913 *c*, p. 713, and 1913 *d*, p. 741.

A good feature of the three papers dealing with this subject is that they set forth in concise fashion the thermodynamic actions that occur.

The attempt to make old processes appear as fundamental cycles, as has been done by Professors Hirshfeld and Ellenwood, may give misleading impressions. For instance, Fig. 20 (p. 684) in their paper makes it appear that the efficiency is less when regeneration is employed, which is by no means the case. This confusion is due to referring efficiency ratios to these processes as though they were fundamental cycles. The matter is finally made clear in their Fig. 30 (p. 703), where heat consumption is plotted against pressure for these various processes. Considerable trouble would be avoided if thermal efficiency also were plotted in Fig. 20.

In reality, the cycles set up by Professors Hirshfeld and Ellenwood are not new kinds of heat cycles, but merely a series of Rankine cycles. Steam supplied to a turbine, condensed in a feedwater heater at any pressure, and then returned to the boiler, operates on a Rankine cycle. Similarly, steam delivered to a prime mover, expanded therein, and exhausted to a reheater is operating on a portion of a Rankine cycle, as does any non-condensing steam engine. I have preferred to consider these various processes as a series of Rankine cycles and see no good purpose for setting up new kinds of cycles and regarding them as fundamental ones. Exceptions to this, of course, are the hypothetical processes described in the paper in which isothermal expansions are employed by adding heat during a portion of the expansion.

I was for some time at a loss to comprehend the meaning of the term "end point" in Professor Wohlenberg's paper. Had he named this "the steam state (or quality) after adiabatic expansion," it would have been clear to every one.

The comparisons in Professors Hirshfeld and Ellenwood's paper have been based on feedwater not heated above the temperature of saturation within the turbine. There is no reason for not carrying this heating into the superheated field except a possible moderate increase in cost of heaters. These authors in their analysis, when they introduce reheating to the cycle, have reduced the amount of regeneration, thereby penalizing the regenerative principle, which is unnecessary and makes the comparison unfair. The general conclusions would have been different had the temperature of feed heating been the same in the various comparisons whether or not reheating were employed.

Messrs. Brown and Drewry in their paper give an interesting method of computing performances where regenerative principles are involved. They, however, propose a deviation of their line on the Mollier diagram when the high-pressure element is of the impulse form, giving as their reason the inherent lower efficiency of this element. This is not necessarily the case. If such an allowance is necessary, it seems that it should similarly be made

for the reaction turbine, for it suffers disability at the high-pressure portion of the turbine because of increased leakage where the steam volumes and blade areas are small. The question of the efficiency of the various-stage elements cannot be generalized but must depend upon the particular design.

Sanford A. Moss.[1] The papers on regeneration, as well as the previous literature, clearly forecast that large power plants should heat feedwater by regeneration, that is, by bleeding from turbine stages. Information is scanty regarding the situation for small power plants. There are many plants using turbines of from 500 to 6000 kw., and it is of course desirable that these plants should be operated with maximum economy, due account being taken of first cost. The exhaustive detail of the three papers tends to indicate that the matter of feedwater heating by bleeding is one of great complexity, and that a great deal of computation would be required to indicate what would be the best situation for a small power plant. However, the following conclusions for small plants seem possible without exhaustive computation, and the writer would inquire if the authors of the papers agree with his conclusions.

The usual small power plant uses auxiliaries driven by non-condensing turbines which exhaust into an open or closed feedwater heater, either with or without an economizer, and never with reheating. Such a plant could substitute motor-driven auxiliaries for many of the non-condensing turbines, and one or more feedwater heaters supplied by bleeding from the turbine instead of the feedwater heater supplied with exhaust steam. There would almost certainly be a decrease of plant investment by this plan. The motor drives would cost appreciably less than the non-condensing turbines. The feedwater heaters using bleeding would probably cost more, but not enough more to wipe out the gain due to motor-driven auxiliaries. Of course, it would not be possible to drive all of the auxiliaries by motors because of the necessity of provision for breakdown and emergency conditions. The substitution of stage heaters for non-condensing heaters will probably decrease maintenance. The many computations make it certain that there will be a small gain in the thermal efficiency of the small plant, and this would be added to the gain due to decreased first cost. It appears, therefore, that a small plant should heat feedwater by bleeding from main turbines. In order that this may come about, feedwater-heater manufacturers must arrange to handle the situation for the small plants. Builders of the turbines are now providing openings in various stages from which steam may be bled.

There is, of course, the necessity for planning the plant layout.

[1] Discussion refers to papers Nos. 1913*b*, page 663, 1913*c*, page 713, and 1913*d*, page 741.

Usually the exhaustive analysis of the present papers is not warranted in laying out a small plant. Nevertheless the feedwater-heating situation requires some attention at the present time. On the other hand, the use of reheaters for a small plant seems to be a much more complicated relation which requires no attention at present.

E. L. Robinson.[1] The author's remarks on his system for parallel feed heating seem to have been too brief. Mr. Rosencrants (p. 766) has noted the best practice according to present standards. But this paper is concerned with the margins between present practice and the ideal. Thus it is inefficient from an ideal viewpoint to make any division of temperature range between the economizer and the extraction heaters. Each should operate throughout the full range. Practical considerations such as cleaning now dictate that the economizer receive warm water, whereas its efficient use from a thermal standpoint requires that it receive as cold water as the turbine-room heaters. Hence the author suggested the parallel feed lines in order that both turbine and boiler-room heaters might utilize the entire temperature range from condenser to boiler. By operating the air preheaters in stages as steam extraction condensers throughout the same temperature range, this scheme retains the practical advantages of the economizer while accomplishing the same ideal as the simpler plan of using only extraction feed heaters and regenerative air preheating from the flue gases.

F. O. Ellenwood.[2] The authors are very grateful to the engineers who have called attention to the inconsistencies contained in Part II of the advance copy of the paper. These inconsistencies were due to errors in the calculations which were of necessity made in much less time than should have been available for the amount of work involved. The calculations for the reheating and regenerative plants involve several new features, among them being Equation [**4**], page 805, containing the ratio, Q_i/Q_a, which has been commonly taken as unity. This is substantially correct for plants operating on the Rankine cycle, but introduces appreciable error for the other cycles as may be seen from Table 5A, page 806. The original tables in Part II have been corrected and three others, Tables 5A, 5B, and 5C, are presented on pp. 806, 807, and 808, in order to give the additional details of the calculations which were not included in the advance copy of the paper. Figs. 21, 22, 23, 24, 25, 30, and 31 have also been corrected to agree with the revised calculations and it is believed that the results will now be found to be consistent, and correct to that degree of accuracy involved in the values given by Fig. 20, which forms the connecting link between the ideal and the real plant.

[1] Closure to discussion on Paper No. 1913 *a*, p. 644.
[2] Closure to discussion on Paper No. 1913 *b* after revision, p. 663.

The curves of Fig. 20 have not been altered, but it is believed that by changing the name from "turbo-generator efficiency" to "engine efficiency of the turbo-generator and heaters" the meaning of the curves has been made clearer.

The method that was chosen to express the probable engine efficiencies as given by Fig. 20 will be found in Tables 2, 3, and 4. These tables were constructed in the endeavor to consider the factors that enter into this question and therefore the authors believed that they could thus determine the engine efficiencies for high pressures better than in any other way, except by the detailed calculation of the plant for each and every pressure. They think that this latter method is the most accurate way to make the estimate for any plant. However, such a method means a most detailed calculation in the factories by the designer and all the others who know the factors of the parts that they are building. Therefore this latter method was not a feasible proposition for the authors at the time they took up this work and could not be considered, and they wish to call especial attention to the fact that it was the whole power plant and not the turbines alone with which they were concerned. If there is anything seriously wrong with Tables 3 and 4 as to the method of getting at the relative degrees of approach toward the ideal cycles, in going to higher pressures, the authors will be glad to have their attention called to it. They do not claim that this method means a high degree of accuracy, but that it is simply a reasonable way of getting at these proportionality factors quickly.

Whether the values given for high pressure by the curves in Fig. 20 are too low or too high for the reheating and regenerative cycle is a debatable point that can only be settled by future tests made available from high-pressure plants. That the low-pressure parts of these curves are probably high enough becomes rather evident by an inspection of Fig. 32, which was not printed in the advance copy of the paper, but which was shown by a lantern slide at the Annual Meeting. If the efficiencies given by Fig. 20 for cycles B and D are too low, then from Fig. 32 it appears that the thermal efficiency of a turbo-generator, including the regenerative feed-heating system, must be greater than 27.5 per cent for an absolute pressure of 200 lb. per sq. in. In other words, a 30,000-kw. turbo-generator operating on the regenerative cycle at that pressure would need less than 12,400 B.t.u. per gross kw-hr. This quantity of heat is measured above the temperature of the feedwater from the high-pressure heater, and the authors believe that such a value is seldom improved upon by our best type of turbines operating under these conditions. For a pressure of 365 lb. per sq. in. abs. and a throttle temperature of 700 deg. fahr., curve D of Fig. 32 shows 11,460 B.t.u. per kw-hr., while the paper of Messrs. Brown and Drewry (p. 713) gives a value of 11,724 B.t.u. per kw-hr. for this same pressure, but for a temperature at the throttle

of 686 deg. fahr., so the agreement at this pressure is very close. It is also worth noting that for this pressure the relative heat consumptions of the Rankine and Regenerative turbo-generators are as follows:

By Brown and Drewry: $\dfrac{\text{Regenerative}}{\text{Rankine}} = \dfrac{11{,}724}{12{,}482} = 0.939$

By Fig. 32 of the authors' paper: $\dfrac{\text{Regenerative}}{\text{Rankine}} = \dfrac{11{,}460}{12{,}200} = 0.939$

From this evidence it appears that the values from Fig. 20 may be found slightly high rather than too low.

If turbine manufacturers or others who have figured the high-pressure regenerative plant believe that better values are obtainable the authors will certainly be glad and they hope, therefore, that their values as given in Fig. 20 for the regenerative are low, as some have suggested. But as far as they have been able to determine those values, it seems to them that they are reasonable ones.

As further evidence of the reasonableness of the values given in Fig. 20 it may be noted that in the revised values of Fig. 30 the same general relations, as given in Fig. 32, are seen to hold for the complete plant, the total auxiliary power being the additional variable factor included in Fig. 30 as compared with Fig. 32.

Regarding the best name to give to the efficiencies expressed by Fig. 20, there will possibly always be some difference of opinion. However, the relations between the ideal and the real plant operating on any one of the various cycles are the important points, which will now be discussed, and certain names will be suggested as possibly better than that originally used in Fig. 20.

Let Q_a = heat supplied in the *actual* plant to the working substance by the economizer, boiler, superheater, and reheater, in B.t.u. per lb. passing the throttle. (In Table 7 this expression is shortened to read "Heat supplied to steam, B.t.u. per lb.")

It should be noted that this quantity expresses the heat supplied above the feedwater temperature to the economizer. It differs from the heat supplied above the feed temperature to the economizer in the ideal case by reason of the losses encountered in the chilling of the condensate in the condenser and the losses of the real reheating and regenerative systems as compared with their corresponding ideals.

Let Q_i = heat supplied in the *ideal* plant to the working substance by the economizer, boiler, superheater and reheater in B.t.u. per pound passing the throttle

E_a = gross energy delivered by the generator of the *real* plant in B.t.u. per pound of working substance passing the throttle

$A.E.$ = energy available for the production of work in the *ideal* plant, in B.t.u. per pound of working substance

e_a = thermal efficiency of the *actual* group of machines consisting of the turbine, generator, condenser, hotwell and regenerative heaters. This group of machines will be referred to as "the group"

e_i = ideal cycle efficiency

e_r = engine efficiency of "the group," or efficiency ratio of "the group."

Then, by definition of these last three terms, it follows that

$$e_r = \frac{e_a}{e_i} \qquad [1]$$

$$e_a = \frac{E_a}{Q_a} \qquad [2]$$

$$e_i = \frac{A.E.}{Q_i} \qquad [3]$$

From [1], [2] and [3] it follows that

$$e_r = \left(\frac{E_a}{A.E.}\right)\left(\frac{Q_i}{Q_a}\right) \qquad [4]$$

and

$$E_a = e_a Q_a = e_i e_r Q_a \qquad [5]$$

The values of e_i are obtained from Part I, and Fig. 20 gives the value of e_r for each of the cycles A, B, D, and E. The values of Q_a and Q_i are different in all cases because of the difference in feedwater temperature, the temperature of water leaving the final heater in the actual case always being lower than that for the corresponding ideal cycle. Furthermore, for the reheating cycles the heat added by the reheater per pound of steam is less in the real case than it is in the ideal, having the same reheating pressure and temperature, because the real turbine delivers the steam to the reheater with a larger specific total heat than does the ideal turbine which permits expansion at constant entropy. The magnitude of this difference, which is shown by columns 15 and 16 of Table 5A, depends upon the internal losses of the turbine and the amount of expansion permitted before the reheating pressure is reached. In Table 5A the total heat of the steam entering the reheater has been estimated by assuming the cumulative stage efficiency of the turbine to be 83 per cent from the throttle pressure to the reheating pressure.

The means used to obtain each value given in Tables 5 and 7 have been inserted at the top of each column, so that no further explanation is needed as to the method of computing them.

TABLE 5A HEAT SUPPLIED IN THE IDEAL AND REAL PLANTS

Cycle	Throttle pressure, lb. per sq. in., abs.	Specific total heat at throttle for temp. of 700 deg. fahr.	Specific Total Heat: Before reheating		Specific Total Heat: After reheating	Heat supplied by reheater, B.t.u. per lb. steam		Temperature of feedwater to economizer, deg. fahr.		Heat of the liquid entering economizer, B.t.u. per lb. water		Heat supplied by economizer, boiler and superheater, B.t.u. per lb. steam		Total supply of heat from economizer, boiler, superheater and reheater, B.t.u per lb. steam	
		Ideal and real	Ideal	Real	Ideal and real	Ideal	Real	Ideal	Real	Ideal	Real	Ideal	Real	Ideal Q_i	Real Q_a
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
A	200	1374	—	—	—	0	0	79	75	47	43	1327	1331	1327	1331
	400	1362	—	—	—	0	0	79	75	47	43	1315	1319	1315	1319
	600	1348	—	—	—	0	0	79	75	47	43	1301	1305	1301	1305
	800	1333	—	—	—	0	0	79	75	47	43	1286	1290	1286	1290
	1000	1317	—	—	—	0	0	79	75	47	43	1270	1274	1270	1274
	1200	1298	—	—	—	0	0	79	75	47	43	1251	1255	1251	1255
B (0.3)	200	1374	1242	1264	1382	140	118	79	75	47	43	1327	1331	1467	1449
	400	1362	1233	1255	1379	146	124	79	75	47	43	1315	1319	1461	1443
	600	1348	1222	1245	1376	154	131	79	75	47	43	1301	1305	1455	1436
	800	1333	1210	1231	1372	162	141	79	75	47	43	1286	1290	1448	1431
	1000	1317	1197	1218	1369	172	151	79	75	47	43	1270	1274	1442	1425
	1200	1298	1183	1202	1365	182	163	79	75	47	43	1251	1255	1433	1418
D	200	1374	—	—	—	0	0	235	188	203	156	1171	1218	1171	1218
	400	1362	—	—	—	0	0	299	252	267	220	1095	1142	1095	1142
	600	1348	—	—	—	0	0	346	302	316	271	1032	1077	1032	1077
	800	1333	—	—	—	0	0	386	344	359	315	974	1018	974	1018
	1000	1317	—	—	—	0	0	422	385	398	358	919	959	919	959
	1200	1298	—	—	—	0	0	457	423	435	398	863	900	863	900
E (0.4)	200	1374	1270	1288	1381	111	93	171	121	139	89	1235	1285	1346	1378
	400	1362	1261	1278	1377	116	99	218	167	185	135	1177	1227	1293	1326
	600	1348	1249	1266	1372	123	106	250	200	218	168	1130	1180	1253	1286
	800	1333	1237	1253	1368	131	115	276	225	245	193	1088	1140	1219	1255
	1000	1317	1222	1239	1362	140	123	299	248	268	216	1049	1101	1189	1224
	1200	1298	1208	1223	1357	149	134	319	268	288	237	1010	1061	1159	1195

The method of obtaining the value given in each column of Table 5A follows:

Column (3): Obtained from steam tables for a throttle temperature of 700 deg. fahr. and pressure as given by (2).
Column (4): Obtained from steam charts for an isentropic expansion from the throttle condition to a pressure of 30 per cent of the throttle pressure for cycle B, and 40 per cent of throttle pressure for cycle E. (See Tables 9 and 14, Appendix No. 1.)
Column (5): = (3) − 0.83 × [(3) − (4)]
Column (6): Obtained from steam tables for temperature of 700 deg. fahr. and pressure of 30 per cent of (2) for cycle B and 40 per cent of (2) for cycle E
Column (7): = (6) − (4)
Column (8): = (6) − (5)
Column (9): Obtained from ideal cycle calculations. (See Tables 9 and 14, Appendix No. 1)
Column (10): Obtained from Tables 3 and 4
Column (11): Obtained from steam tables to agree with (9)
Column (12): Obtained from steam tables to agree with (10)
Column (13): = (3) − (11)
Column (14): = (3) − (12)
Column (15): = (7) + (13)
Column (16): = (8) + (14)

For Table 5B all equations are given at the top of each column.

TABLE 5B HEATING SURFACES OF SUPERHEATER AND REHEATER FOR 30,000 KW. GROSS OUTPUT FROM GENERATOR

Cycle	Estimated boiler pressure, lb. per sq. in., abs.	Specific total heat of saturated steam at boiler pressure	Specific total heat of steam at the throttle for temp. of 700 deg. fahr.	Steam flow, lb. per hr. (From Table 5)	Surface of superheater in sq. ft. $S_s = \frac{W_s(H_c - H_{sat.})}{3700}$	Heat added by reheater, B.t.u. per lb. of steam (From Table 5A)	Reheater surface in sq. ft. $S_r = \frac{W_sQ_r}{3700}$
		$H_{sat.}$	H_c	W_s	S_s	Q_r	S_r
A	210	1199	1374	293,000	13,900	—	0
	417	1202	1362	276,000	11,900	—	0
	841	1193	1333	268,000	10,100	—	0
	1263	1178	1298	273,000	8,850	—	0
B (0.3)	208	1199	1374	257,500	12,200	118	8200
	412	1202	1362	240,000	10,400	124	8050
	819	1193	1333	229,000	8,650	141	8700
	1226	1178	1298	228,000	7,400	163	10000
D	211	1199	1374	308,000	14,500	—	0
	420	1202	1362	298,000	12,900	—	0
	836	1193	1333	310,000	11,700	—	0
	1256	1178	1298	339,000	11,000	—	0
E (0.4)	209	1199	1374	273,000	12,900	93	6850
	414	1202	1362	257,000	11,100	99	6900
	823	1193	1333	250,000	9,450	115	7800
	1232	1178	1298	254,000	8,250	134	9200

TABLE 5C HEATING SURFACES OF AIR HEATER, BOILER, AND ECONOMIZER FOR 30,000 KW. GROSS OUTPUT FROM GENERATOR

(Steam at throttle = 700 deg. fahr. Gas to stack: W_g = 14.2 lb. per lb. of coal; temp., t_2 = 269 deg. fahr. Air to heater: W_a = 13.3 lb. per lb. of coal.)

Cycle	Boiler pressure, lb. per sq. in., abs. p_1	Temperature of gases from boiler to economizer $t_1 = t_{sat.} + 140$ t_1	Temperature of air to furnace (Assumed) t_a	Heat absorbed by air from air heater, in B.t.u. per lb. of coal $Q_a = 0.24\,W_a(t_a - 70) = 3.19\,t_a - 223$ Q_a	Temperature of flue gas from economizer to air heater $t_2 = 0.937t_a + 204$ t_2	Coal per hr., lb. (From Table 7) W_c	Mean temperature difference in air heater $\Delta t_a = \frac{(t_2-t_a)-(t_3-70)}{\log_e[(t_2-t_a)/(t_3-70)]}$ Δt_a	Surface of the air heater in sq. ft. $S_a = \frac{Q_a W_c}{3\Delta t_a}$ S_a	Heat given up by gases in economizer in B.t.u. per lb. of coal $Q_g = 0.24W_g(t_1 - t_2)$ Q_g	Total steam flow, lb. per hr. (From Table 5) W_s	Heat given to feedwater by economizer B.t.u. per lb. of steam $Q_e = \frac{Q_g W_c}{W_s}$ Q_e	Temperature of feedwater entering economizer (From Table 5) t_f	Temperature of feedwater leaving economizer $t_k = Q_e + t_f$ t_k	Specific total heat of saturated steam at boiler pressure $H_{sat.}$	Heat of the liquid at temperature of feed to boiler q_k	Heating surface of the boiler in sq. ft. $S_b = \frac{W_s(H_{sat.} - q_k)}{5360}$ S_b	Mean temperature difference in economizer $\Delta t_e = \frac{(t_1 - t_k) - (t_2 - t_f)}{\log_e[(t_1 - t_k)/(t_2 - t_f)]}$ Δt_e	Economizer surface in sq. ft. $S_e = \frac{W_c Q_g}{5\Delta t_e}$ S_e
	210	526	70	0	269	37,700	—	0	875	293,000	112	75	187	1199	155	57,000	260	25,400
	417	589	70	0	269	35,200	—	0	1090	276,000	139	75	214	1202	182	52,400	275	27,900
	841	664	70	0	269	33,500	—	0	1349	268,000	169	75	244	1193	212	49,000	293	30,900
	1263	714	70	0	269	33,200	—	0	1518	273,000	184	75	259	1178	227	48,400	307	32,900
B (0.3)	208	525	70	0	269	36,100	—	0	875	257,500	123	75	198	1199	166	49,600	255	24,800
	412	588	70	0	269	33,500	—	0	1089	240,000	152	75	227	1202	195	45,000	269	27,100
	819	661	70	0	269	31,700	—	0	1337	229,000	185	75	260	1193	228	41,200	285	29,800
	1226	710	70	0	269	31,400	—	0	1502	228,000	207	75	282	1178	251	39,400	295	31,900
	211	526	128	186	324	36,300	198	11,400	688	308,000	81	188	269	1199	238	55,200	190	26,300
	420	590	170	319	363	33,100	196	18,000	774	298,000	86	252	338	1202	309	49,600	172	29,800
	836	664	240	543	429	30,600	194	28,500	802	310,000	79	344	423	1193	398	45,800	150	32,800
	1256	713	300	734	485	29,500	192	37,500	776	339,000	68	423	491	1178	473	44,600	126	36,500
E (0.4)	209	526	70	0	269	36,500	—	0	875	273,000	117	121	238	1199	206	50,500	211	30,200
	414	588	70	0	269	33,100	—	0	1090	257,000	140	167	307	1202	276	44,400	177	40,800
	823	662	140	223	335	30,400	197	11,500	1113	250,000	135	225	360	1193	331	40,200	190	35,600
	1232	711	180	352	372	29,500	196	17,700	1155	254,000	134	268	402	1178	376	37,900	188	36,300

For Table 5C the only equation which may require derivation is the one for t_2, the temperature of the flue gas from the economizer entering the air heater. Using the notation given in this table and noting that the temperature of the gas to stack is assumed to be 269 deg. fahr., then it follows, for no loss by radiation or conduction from the air heater, that

Heat absorbed by the air = Heat given up by the gas

or

$$W_a c_p (t_a - 70) = W_g c_{p'} (t_2 - 269)$$

The value of c_p, the specific heat of the air, is nearly the same as that of the gas $c_{p'}$, so that if they are considered equal, then for $W_a = 13.3$ and $W_g = 14.2$, the above equation gives $t_2 = 0.937t_a + 204$.

Attention should also be called to the fact that in Table 5C the heating surface of the boiler, S_b, does not include the superheating surface, S_s, which is given in Table 5B.

The constant, 5360, or 1.6×3350, used in Table 5C represents the rate of heat transfer in B.t.u. per hr. per sq. ft. of heating surface for a rating of 160 per cent as used in this paper.

The rating of a boiler has long been a fair subject for debate and it is not intended, at this time, to discuss that subject in general, but it is the belief of the authors that the fairest method of making a comparison of the heating surfaces needed in plants using such widely different pressures as are involved in this paper, is to keep the superheating surface distinct from the wet heating surfaces.

Regarding the reheating cycles and the high efficiencies which result the authors do not object in the least to the statements that have been made regarding those high efficiencies. They are heartily in agreement with them, but it is because of the complexity that is introduced by virtue of this reheating proposition — at least it seems to them that way — that they have chosen the regenerative cycle as the final recommendation at the present time even though its economy from a thermal standpoint may be slightly less.

Attention has been called to the advantage of using separators in the turbine in order that some of the water might be extracted as the steam passes through the turbine. The authors do not claim to know whether that problem can be solved or not. They do claim, however, that it appears to be such an attractive proposition that they believe turbine designers ought to make an extra effort to see if more cannot be done than is known about it at the present time. If they have tried it, and tried it very faithfully, they should give details as to why it failed.

The bleeding of superheated steam, as mentioned by several engineers, involves so many factors that the authors did not have time to consider it in this paper. However, they believe that it is perfectly logical to make the comparison of the ideal cycle with

the actual plant in which bleeding begins in both cases as soon as the saturation curve is reached. They also believe it would be logical to make a comparison when the bleeding begins, in both the ideal and real cases, with superheated steam at some pressure common to each case. Both types of actual plants are in use today. The authors agree with those who believe that the regenerative cycles with a throttle temperature of 800 deg. fahr. would probably show much better results if the bleeding should begin in the superheated region, but that is not the case considered for any cycle in this paper.

Professor Wohlenberg has correctly inferred that the low engine efficiencies assigned to the regenerative cycles follow from the fact that actual equipment, with but few extraction feedwater heaters, is compared with ideal equipment with an infinite number of heaters. The engine efficiencies therefore include, as he points out, imperfections in the actual cycles as well as imperfections in the application of these cycles in real machines.

With respect to the superheater and reheater heat-transfer rates, which Professor Wohlenberg criticizes, the authors may cite the results obtained by Mr. P. W. Thompson.[1]

In that test the purely convection type of superheater absorbed heat at rates varying from about 2000 to 7800 B.t.u. per sq. ft. per hour for boiler ratings ranging from 85 to 270 per cent. Similar results have also been obtained elsewhere, but so much depends upon the design and location of the superheater that it is useless to argue this point.

Mr. Broido states that Fig. 19 indicates that at the higher pressures the reheating-regenerative cycle is less efficient than the regenerative one. In reply the authors desire to call his attention to the fact that this figure does not represent efficiencies at all, but the available energy that may be obtained, from the steam in passing through an ideal turbine, per unit volume of steam passing to the condenser. In other words, the relative values shown by Fig. 19 are roughly inversely proportional to the relative amounts of condenser surface, as may be seen from a comparison of Figs. 19 and 21.

Regarding high-pressure turbines, Mr. Alden said that decreased costs might result from going to high pressures and that higher values of the efficiencies might be obtained than those given in Fig. 20, for the regenerative cycle. If that is so, it will strengthen the author's final conclusions. If the high-pressure plant costs less than they have estimated, so much the better. If the efficiency of the regenerative cycle is higher than they have estimated, again so much the better. So they hope that Mr. Alden is right in every particular.

[1] Tests of a Type W Stirling Boiler at the Connors Creek Power House of The Detroit Edison Company. Trans. A.S.M.E., vol. 44 (1922), p. 1005.

Mr. Landis has described an interesting and valuable graphical representation of the processes involved in the regenerative cycle, perhaps, as he himself states, not well adapted to convenient and quick representation for purposes of computation, but certainly very satisfactory for displaying the physical processes in a readily comprehensible manner. The authors might suggest that his figures would be even clearer if the temperature-entropy-mass volumes, that were removed to represent steam extracted during expansion, were drawn at the front of the diagram rather than at the back. They cannot agree with him, however, when he states that the authors' Fig. 4 is not a temperature-entropy diagram for an ideal regenerative cycle. It is believed that Par. 13 states the conditions for which Fig. 4 is drawn. There are other conventional temperature-entropy diagrams which may also be used for this cycle. Each one possesses certain points of value over the others.

Regarding the point brought up by Mr. Drewry (p. 812) as to whether supersaturation, or excessive moisture, is the cause of the reduced efficiencies in the low-pressure stages of turbines, it is of course impossible to be sure. However, the authors feel that the internal reheating in a turbine using wet steam is certainly much greater than in one using superheated steam. Whether the term "friction" should be used to express this fact may be a debatable point, but when one observes the erosion of turbine blades which have been used for a few years with wet steam impinging upon them, it makes one believe that the moisture has certainly been a big factor in producing the worn blades. This worn condition of the blades was surely produced by an expenditure of energy which came from the steam, and whether or not this loss should be spoken of as "friction" is immaterial.

Regarding the names of the cycles, Mr. Hodgkinson said that he would prefer to call them all Rankines. With that statement the authors disagree as they believe that we name things for the purpose of enabling us to talk about them, and if things are different they should have different names. Moreover, it is hard to change technical definitions. Referring to Fig. 20, most of those who discussed the paper orally evidently think of it in terms of "efficiency ratio" and it would be difficult, indeed, to get the average engineer to think of cycles (which, strictly speaking, are not Rankines) as Rankines if a number of things that modify that cycle were to be changed so that Rankine himself would never have recognized it.

Mr. Rosencrants points out that higher regenerative-cycle efficiencies could be obtained by continuing the extraction feed-water heating above the point at which the adiabatic expansion line intersects the saturation curve. This is unquestionable. The authors limited their studies arbitrarily and consistently, so that all results might be properly comparable. They have not thereby

expressed it as their opinion that this limit is desirable in actual practice.

If Mr. Rosencrants is correct in assuming that the turbine can utilize steam at as high temperatures as present-day piping and superheaters can supply it, then of course he is correct in stating that a loss results from using superheated steam for reheating expanded steam. This is a question of machine and plant design and construction, on which much light will be thrown by developments already in progress.

M. K. Drewry.[1] Professor Christie is right in calling attention to the approximation occurring in Item 43. Although the authors only intended that value and also Items 41 and 42 as approximate design values, they inadvertently did not label them as such. Item 43 may be obtained accurately by multiplying Item 39 or Item 40, the steam rates when extracting, by Item 1, the output. As representing manufacturers of turbines, the authors have been primarily interested in economies, and have therefore overlooked the occasional need of commensurate accuracy in design values.

The pipe-loss drop of 5 per cent and extraction-nozzle decrease in pressure of 2 per cent are arbitrary, unless specific installations are considered. It was the intention of the authors to indicate all corrections, however small or unimportant, so that allowance for them would not be overlooked in actual application of the method. One loss omitted, however, consists of the steam vapor discharged in the air vents from one heater to the other, if such occurs.

In substantiation of Professor Christie's discussion of the use of superheated steam in extraction heaters, the results of some computations as to the loss caused by desuperheating that steam are submitted. They show that in a typical non-reheating cycle the loss is only one-twelfth of one per cent, though in a reheating cycle it amounts to six-tenths of one per cent, an amount that does not forestall the use of the desuperheater.

If experience shows desuperheaters to be more commercially desirable than special heaters, they should be applied in any case, for the losses cited above are much less than those due to limitation of final feedwater temperature.

Another point brought out in the discussion was that turbine efficiencies in the higher-pressure sections in certain types of machines were not as low as the authors had represented them. In preparing the paper it was necessary to give a workable hypothesis. The amount of information in this country regarding impulse turbine efficiency is not great. Stodola's book, however, gives efficiencies for these elements of about 60 per cent, and these curves for the other turbines were inserted in the paper as an approximation, for the reason that nothing better was available. The au-

[1] Closure to discussion on Paper No. 1913 c, p. 713.

thors have since learned that 70 per cent was more nearly correct for this type of machine than the 66 per cent given in the paper. In this connection, they have in Fig. 5 represented the straight reaction machine as having an efficiency of 81½ per cent in the high-pressure section. Actually reaction turbines as tested have shown higher efficiencies than this.

The question of steam-driven auxiliaries is interesting in this connection. The authors, in their paper, made a study of the steam-jet air pump, taken from a typical installation at present in operation. The study showed that, if considered in its relation to the overall plant efficiency, 0.80 per cent of the total steam was fed to the auxiliary, and that 0.72 per cent was chargeable against the unit with no credit for feedwater heating. It should be appreciated that the true function of a power plant is to deliver power, and not to heat feedwater. The steam-jet air pump has an efficiency based on the work done of perhaps not more than one or two per cent, while the piston air pump has a thermal efficiency of from 10 to 20 times as much. There seems to be no question, aside from liability considerations, but what the electrically-driven auxiliary should displace the steam-driven one.

One of the papers presented expressed considerable confidence as to the possibilities of bettering turbine efficiencies by extracting surplus moisture from the lower pressure stages. Though an extremely debatable subject, since it involves the elusive subject of supersaturation, it appears without basis when considered from the following viewpoints.

First, some proponents of the practice suggest that "friction" of the moist steam is the cause of the loss, claiming a greater loss of frictional pressure drop with moist than with dry steam. In the light of many experiments showing that the drop in the head necessary to force a given fluid through passages at high velocities is independent of the nature of those fluids — that at high velocities water and molasses pass through restrictions with equal pressure drops — this theory appears untenable.

Power loss due to moisture could occur by alternate inefficient acceleration and stopping of separated moisture in the forms of "slugs" of water. The theory of supersaturation of steam indicates the presence of moisture drops on the internal turbine casing, but that these few drops suspended in the moving fluid should cause a considerable power loss seems improbable.

Since supersaturation occurs only in the saturated field, one is led to attribute some of the 10 per cent difference in efficiency in the superheat and saturated fields to that phenomenon. That the degree of such loss is a function of the moisture cannot be said with assurance. In fact, some writers question the presence of the loss past the Wilson line, which roughly follows the 95 per cent quality line of the Mollier chart. However, since opinion does say that the loss probably does occur immediately upon crossing the

saturation line, it is apparent that *reheating* in external heaters, not mere moisture extraction from the casing, would be the only apparent solution.

It is felt that the greatest percentage of the observed 10 per cent greater efficiency in superheated than in saturated fields, as occurs in straight reaction-type machines, is due to the wide difference in internal reheating in these fields. A glance at the comparative non-parallelism of the pressure lines in the two fields, as shown on the Mollier chart, will illustrate this point.

W. J. Wohlenberg.[1] Professor Christie mentions an optimum or best pressure of 1000 lb. It is important to note that this optimum exists when the station is designed for a given steam-generator efficiency. For a given steam generator, as shown by curves D to H, Fig. 13 (p. 750), this optimum is considerably lower.

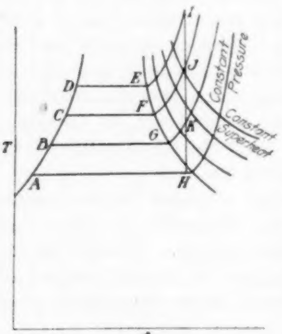

Fig. 12

The data on the end-point curve submitted by Professor Christie seem to bear out the general trend of the relation. It should be remembered, however, that each turbine type may have its own end-point curve.

Mr. Broido has presented some very valuable information on reheating-cycle and high-pressure-boiler developments in Europe. The interstage reheating cycle in which reheat is absorbed from high-pressure steam certainly presents a practical solution of such a problem which may bear further investigation. Of course, by this method the heat is indirectly withdrawn from the flue gases, and it is apparent that isothermal reheating may be approached. Furthermore, the definite temperature control possible with this method is of great advantage, and surface requirements in reheaters should be greatly reduced.

Mr. Orrok mentions the fact that the internal-machine-efficiency coefficients probably should have been taken for machines designed for the conditions, rather than extrapolated to the high-pressure and reheat field from data available from a given machine. This point is important, but as accurate data for high-pressure reheating machines are not available it seemed that for relative values the method used would yield better results than attempted extrapolation of doubtful coefficient variations. At any rate, the high-pressure corrections are probably on the safe side, so that gains in economy may be expected up to the optimum-pressure values

[1] Closure to discussion on paper No. 1913*d*, p. 741.

indicated. Whether or not the divergence of efficiency values for reheating and for other machines is too great may be determined only after more experimental data are available. By means of the end-point curve the same variation of machine efficiency with conditions is made the basis of determination for all cycles, and, as is later shown, the validity of such a method may be demonstrated.

The End-Point Curve and What it Assumes. Mr. Helander emphasized the fact that figures taken from the curve in Fig. 10 might not be applied with theoretical accuracy to reheating-cycle machines if the curve was constructed from test data for Rankine-cycle machines. It is the purpose of this analysis to expose, so far as is possible, the structure on which such a relationship is

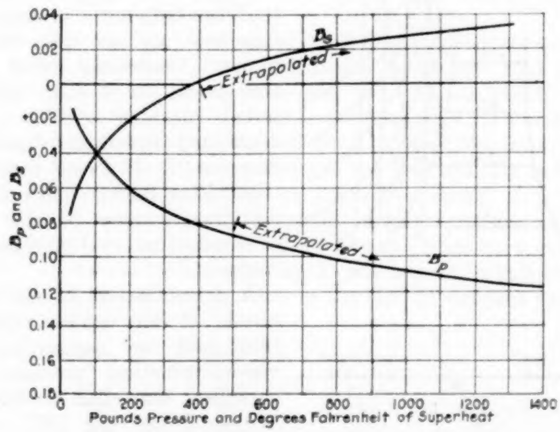

FIG. 13 BAUMAN COEFFICIENTS OF 1921

based, in order that engineers may know more exactly how it may be applied. It shows that for the field considered, the methods used involve errors so small as to be negligible with the present knowledge of the subject, but that when applying the method by extrapolation from Rankine-cycle to reheating machines the errors involve the type of high-pressure elements in the machine under consideration.

In the application of Fig. 10 to Rankine-cycle machines, from the test data of which type the curve was constructed, the relationship assumed as to such cycles is illustrated by reference to the accompanying temperature-entropy diagram, Fig. 12. In what follows the term efficiency is used to designate internal machine efficiency. Referring first to Fig. 10, it is noted that for a given back pressure and adiabatic end point there is a given machine efficiency. Referring next to Fig. 12 (p. 814), we find that for a given back pressure $A \doteq H$ any number of Rankine cycles such as $ABGKH$,

ACFJH, etc., may be traced through for the same end point *H*. It follows that the end-point curve assumes for a given back pressure such as $A - H$ that adiabatic lines such as $H - I$ are also virtually lines of constant efficiency within the limits of the problem investigated.

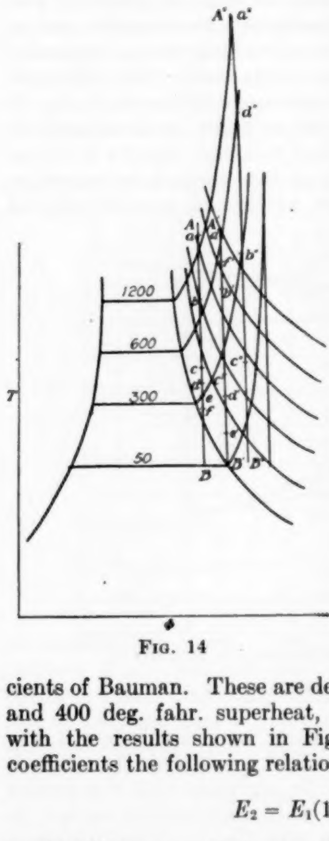

Fig. 14

Let us see, first, if such a relation appears possible, and, second, if it can be verified within reasonable limits for the range of conditions investigated. Comparing the Rankine cycles *ADEIH* and *ACFJH*, it is apparent that for the latter both initial pressure and superheat are less than for the former. Decreasing initial pressure tends to increase the internal machine efficiency, and decreasing superheat tends to decrease it. It seems, therefore, that within limits, such adiabatics virtually may be also lines of constant internal machine efficiency.

A determination of the magnitude of any errors resulting from such an assumption involves practical pressure and superheat correction coefficients. We shall use the 1921 coefficients of Bauman. These are developed only up to 500 lb. pressure and 400 deg. fahr. superheat, but the curves may be extended with the results shown in Fig. 13 (p. 815). In applying the coefficients the following relation is used:

$$E_2 = E_1(1 + B_p + B_s)$$

in which E_2 represents the corrected efficiency obtained by applying coefficients B_p and B_s from the curves. In the investigation, a field such as is illustrated by adiabatics AB, $A'B'$, $A''B''$, etc., in Fig. 14 above will be covered. The following adiabatic is first considered:

Point	a	b	c	d	e	f
Pressure, lb.	1200	600	500	400	300	240
Temperature, deg. fahr.	800	640	575	522	460	410

The following net corrections result:

	a to b	a to c	a to d	a to e	a to f
$B_p + B_s$ =	0.007 +	0.002 +	0.001 +	0.008 −	0.028 −
Supheater (deg. fahr.) at	$b = 154$	$c = 108$	$d = 77$	$e = 43$	$f = 12$

It appears that until the superheat is well below 50 deg. fahr· the net correction is very small, in fact one-half of one per cent or less. If the correction factors $(B_p + B_s)$ are plotted against pressure a haphazard sort of curve is discovered, but the absolute value of the correction is inconsiderable until the low superheat is reached.

A second adiabatic to the right of the first one yields the following data:

Point	Pressure, lb.	Temp., deg. fahr.	Superheat, deg. fahr.	Correction $B_p + B_s$
a'	600	800	314	
b'	300	615	198	a' to b' = 0.006 +
c'	200	520	138	a' to c' = 0.007 +
d'	150	460	102	a' to d' = 0.009 +
e'	100	373	45	a' to e' = 0.000
f'	400	690	(e') 45	f' to e' = 0.007 −

The correction factor is again small and inconsistencies appear if a curve is plotted. Now it is logical to assume that no such irregular variations in efficiency will occur as the initial or high point of the cycle is moved along an adiabatic line. The conclusion is that within the range of condidions of this problem the end-point curve yields efficiency values that will be more accurate as to trend than any coefficients arrived at through a series of correction factors. Of course, for initial conditions of very low superheat and pressure it will be necessary to have an additional end-point curve. However, such conditions do not appear in this problem.

Finally, a comparison is made in the field of extreme superheat as shown below:

Point	Pressure, lb.	Temp., deg. fahr.	Superheat, deg. fahr.	Correction $B_p + B_s$
a''	1200	1240	672	
b''	300	800	383	a'' to b'' = 0.017 +
c''	100	520	192	a'' to c'' = 0.021 +
d''	600	1007	521	d'' to c'' = 0.003 +

It is to be noted that for the first two cases above the correction is in the vicinity of 2 per cent, but that in the third case it is less than 0.3 per cent. It appears, therefore, that for the very high, as well as for the very low, superheat a special end-point curve should be constructed. However, the coefficients B_p and B_s have been extrapolated a great distance to obtain these results, and probably little faith may be placed in the magnitude of the net correction.

To sum up, we may now say that for the principal field of initial steam conditions of this problem the end-point curve probably yields for the Rankine-cycle machines efficiency values more consistent and of better relative trend than would be those obtained by a series of correction coefficients, and, furthermore, the error in absolute values certainly will be within one per cent. The investigation brings out three important conditions concerning the variation of internal machine efficiency as the high point of the cycle is moved along an adiabatic line. First, within the usual operating range, adiabatics are virtually constant-efficiency lines. Second,

in the very-low-superheat field the efficiency decreases along decreasing adiabatics because of the preponderant influence of decreasing superheat. Third, in the very-high-superheat field the efficiency decreases along adiabatic lines for increasing conditions because of the preponderant influence on efficiency of increasing pressures.

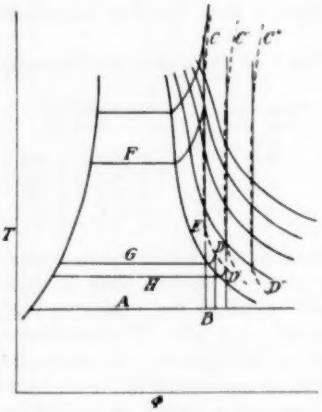

Fig. 15 Constant-Efficiency Lines[1]

The net result is illustrated by the constant-efficiency lines shown dotted in Fig. 15. The same lines may be represented in a Mollier chart. In constructing the lines it should be remembered that the end-point relation assumes a constant back pressure, and that, therefore, the curves assume the same condition to be true. It is now important to note that such a picture of the probable trend of machine efficiencies may be of considerable assistance in determining actual trends and values.

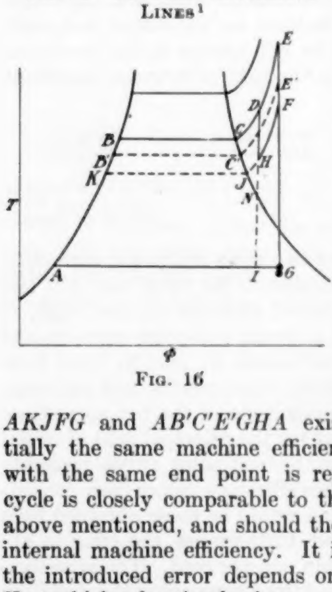

Fig. 16

Having demonstrated the validity of this method for the Rankine cycle, it now becomes necessary to investigate further in order to discover the magnitude of errors involved when applying the end-point relation to reheating machines.

End-Point Curve Applied to Reheating Cycles. Referring to Fig. 16, it is seen that for a given end-point G any number of Rankine cycles such as $ABCEG$, $AKJFG$ and $AB'C'E'GHA$ exist, and they will have substantially the same machine efficiency. A one-stage reheating cycle with the same end point is represented by $ABCDHFG$. This cycle is closely comparable to the first and third Rankine cycles above mentioned, and should therefore have very nearly the same internal machine efficiency. It is obvious that the magnitude of the introduced error depends on the relative height of the point H at which reheating begins.

[1] (Dotted lines show probable trend of constant internal machine efficiencies. $C - D$ = constant internal machine efficiency curve for back pressure A. All Rankine cycles such as $(H - A)$, $(G - A)$, and $(F - A)$ have same machine efficiency. Above point E all such cycles have practically same end-point B.)

In order to evaluate the efficiency of the reheating cycle in an apparently logical manner, we shall divide the cycle into two parts as suggested by Mr. Helander, the Rankine cycle *AKJFG* having an efficiency equal to that found from the end-point curve, and a second cycle *KBCDHJK* whose efficiency may be either higher or lower. The latter cycle will usually be contained within the superheated region, under which condition the deciding factor as to plus or minus error in efficiency may be determined by the type of high-pressure elements in the machine under consideration. It follows that the latter condition will have a bearing on the value and required amount of reheating. The above cycles may be designated as $KBCDHJK = B$, and $AKJFGA = C$.

The corrected mean efficiency may now be stated in terms of

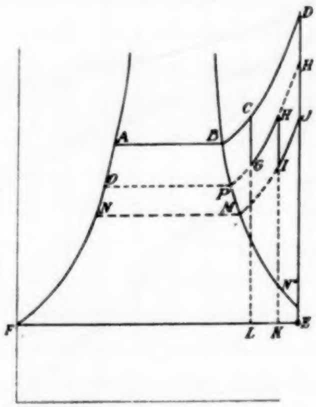

FIG. 17

a weighted average in which the weights are taken equal to the expected heat drops during expansion in each cycle. These are: for cycle B, $\Delta i_B \cdot E_B$; and for cycle C, $\Delta i_C \cdot E_C$. In these expressions Δi_B and Δi_C denote the adiabatic heat drops of the ideal cycles. The corrected mean efficiency is represented by

$$E_m = \frac{\Delta i_B \cdot E^2{}_B + \Delta i_C \cdot E^2{}_C}{\Delta i_B \cdot E_B + \Delta i_C \cdot E_C} \quad \ldots\ldots \quad [1]$$

It is to be noted that the efficiency E_m as here developed will apply to that reheating cycle which has the points D and F in exactly the same position as those occupied by these points in that ideal reheating cycle for which the ideal thermal efficiency is evaluated. The product of the two efficiencies is thus the actual thermal efficiency of a reheating cycle with the fixed points D and F. If the ideal performance is to be related to the actual performance of a distorted actual cycle in which the point F has been displaced to the right of the position it occupies in the ideal cycle,

the final result must be arrived at in a somewhat different manner, which will be shown later.

Before expression [1] may be evaluated it is necessary to determine the efficiency E_B. This may be approximated in the following manner. Find E_{DI} for the Rankine cycle of which B is the top, by means of the end-point curve for end-point I. Next determine the point N at which the expansion line DI cuts the saturation line. Then by applying a practical relation between efficiency and the

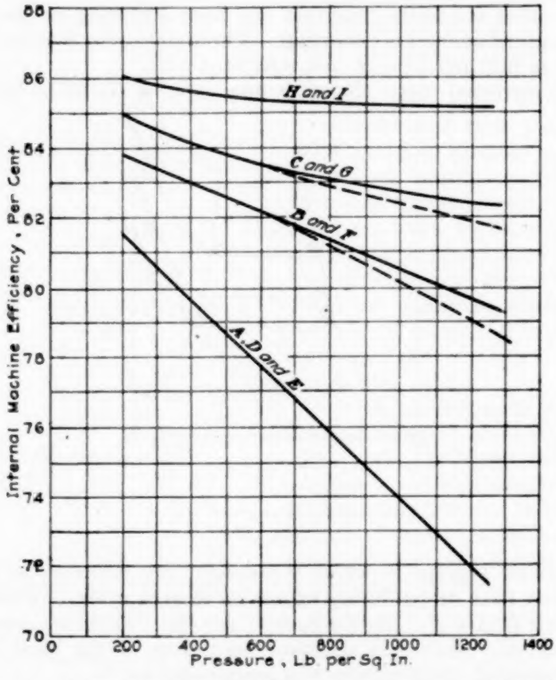

Fig. 18 Internal Machine Efficiency, Parsons 30,000-Kw. Compound Turbine

(Back pressure = 1 in. Hg; H and I = 1000 deg. fahr. steam temp.; H = 2-stage reheating; I = 2-stage reheating plus 2-stage bleeding; C, G, B, F, A, D, E = 750 deg. fahr. steam temp.; C = 2-stage reheating; G = 2-stage reheating plus 2-stage bleeding; B = 1-stage reheating; F = 1-stage reheating plus 1- or 2-stage bleeding; A, D, and E = Rankine and regenerative machines.)

superheated and expansion ranges, expression [2] may be evaluated for either E_{DN} or E_{NI}.

$$\frac{\Delta i_{DN}\cdot E^2_{DN} + \Delta i_{NI}\cdot E^2_{NI}}{\Delta i_{DN}\cdot E_{DN} + \Delta i_{NI}\cdot E_{NI}} = E_{DI} \quad . \quad . \quad . \quad . \quad . \quad [2]$$

It may be argued that greater accuracy is obtainable if the

actual heat drops $\Delta i'_{DN}$ and $\Delta i'_{NI}$ in the superheated and saturated fields are found and the expression is used in the form

$$\frac{\Delta i'_{DN} \cdot E_{DN} + \Delta i'_{NI} \cdot E_{NI}}{\Delta i'_{DN} + \Delta i'_{NI}} = E_{DI} \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{2a}]$$

That the two expressions will yield slightly different results is apparent when noting that [**2a**] gives greater weight to E_{DN} and less to E_{NI} as the efficiency in the superheated field decreases. This is as it should be, whereas in [**2**] the weights are entirely determined by the point at which the adiabatic line cuts the saturation curve.

For Parsons turbines, Messrs. Brown and Drewry point out in their paper[1] that E_{DN} is 10 per cent greater than E_{NI}. The latter term therefore may be replaced in [**2**] and [**2a**] by $\left(\frac{E_{DN}}{1.1}\right)$, whence E_{DN} may be determined. But E_{DN} is practically equal to E_B if Δi_B is all in the superheat field, whence expression [**1**] may be solved for E_m. By using [**2**], for convenience, the following results are obtained in the solution:

Initial pressure, lb. per sq. in.	300	1200
Efficiency from end-point curve, per cent	83.4	79.6
Corrected value of E_m from [**1**], per cent	83.45	78.8

The two-stage reheating cycle is shown in Fig. 17 (p. 819). A comparable Rankine cycle has its high point at H'. The reheating cycle may be divided into the following part cycles: $OABCGPO = B$; $OPHIMNO = C$; $NMIJEFN = D$. Then

$$E_m = \frac{\Delta i_B \cdot E^2_B + \Delta i_C E^2_C + \Delta i_D \cdot E^2_D}{\Delta i_B \cdot E_B + \Delta i_C \cdot E_C + \Delta i_D \cdot E_D} \quad . \quad . \quad . \quad . \quad [\mathbf{3}]$$

in which the values of E_B and E_C are found in a manner similar to that previously outlined for finding E_B in the one-stage cycle. The following results are obtained:

Initial pressure, lb. per sq. in.	300	1200
Efficiency from end-point curve, per cent	84.5	82.2
Corrected value of E_m from [**3**], per cent	84.4	81.3

It is to be noted that in both of the cases investigated the maximum difference is less than one per cent, which is probably within the error of necessary assumptions.

In Fig. 18 (p. 820) the end-point efficiency values are indicated by solid lines and the corrected values E_m by dotted lines. It is apparent that below optimum pressures the divergence is exceedingly small. For the type of machine investigated it appears, therefore, that the trends and relative positions are very nearly correct.

In working out the above corrected values of E_m, it has been

[1] Economy Characteristics of Stage Feedwater Heating by Extraction. See page 713.

assumed as before noted that the point F, Fig. 16, is that of the ideal cycle, whereas actually it might be displaced to the right as shown by F' in Fig. 19. This condition is caused by the reduction in optimum reheating pressures due to internal machine losses such as friction and leakage. For such cases the end-point efficiency values should be taken as those corresponding to the displaced end-point, H'. The actual and ideal cycles are represented respectively by areas $ABCDE'F'G'$ and $ABCDEFG$. It is obvious that the end-point efficiency for H' may not be multiplied into the thermal efficiency of the above ideal cycle in order to obtain the thermal efficiency of the actual cycle. The ideal heat drops during

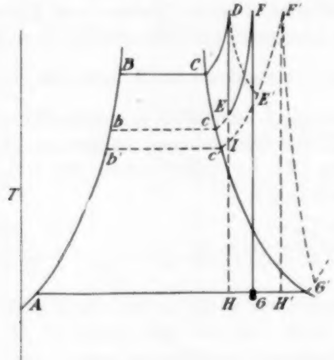

Fig. 19

the expansion of the actual cycle are now those on the adiabatics DI and $F'H'$ in place of those on the adiabatics DE and FG of the ideal cycle. It is apparent for this case that the problem is one of relating the ideal and real performances of a modified cycle rather than one of evaluating the work expected by discovering the fraction of available energy in an ideal cycle which may be converted into work. The latter method has been used in the paper. In order to solve the former it is merely necessary to multiply the sum of the adiabatic heat drops from D to I and from F' to H' by the end-point efficiency for H'. This product represents expected work, which may then be related to other quantities involved.

If a greater refinement is to be resorted to in the evaluation of the machine efficiency for this case it may be found, in a manner similar to that already outlined, from the relation

$$E_m = \frac{\Delta i_{DI} \cdot E^2_{DI} + \Delta i_{F'H'} \cdot E^2_{F'H'}}{\Delta i_{DI} \cdot E_{DI} + \Delta i_{F'H'} \cdot E_{F'H'}} \quad . \quad . \quad . \quad . \quad . \quad [4]$$

in which the subscripts refer to points shown in Fig. 19. Evaluation of E_{DI} involves, as before, the relation between efficiencies in the high- and low-pressure sections of the turbine under consideration.

The question arises as to the degree of error involved in the direct application of the end-point relation to reheating machines in which a difference exists in the type of high- and low-pressure sections. Examples of such machines are seen in the Westinghouse combination and in the Curtis-Rateau machines. In both cases a velocity compounding wheel is used in the high-pressure section, which, relatively speaking, has a low stage efficiency. For such cases the end-point efficiency taken from a Rankine-cycle machine curve may be low, and Equation [**4**] or its equivalent is recommended.

Once reheating machines are in use, the whole problem will, of course, be simplified, as end-point curves may be constructed directly from then available data for such machines.

No. 1914

MEASUREMENT OF MANAGEMENT

By Joseph W. Roe,[1] New York, N.Y.

Member of the Society

In this paper the author discusses the need of measurement and the characteristics necessary in any method of measuring management, and outlines a method for comparing managerial performance in different plants or variations thereof in a single plant.

THE purpose of this paper is to outline the problem and to suggest a method for evaluating management which will be trustworthy, practically useful, and as free as possible from the element of personal opinion.

2 *Is Management Measurable?* It is desirable to face this question squarely in view of the frequency with which we encounter one or the other of two answers to it. The first is, "Of course it is, and the balance sheet and profits are the measure." This is too easy to be true. Profits are usually an indication of good management, not a quantitative measure of it. They depend on conditions beyond the control of the management, as well as on those within its control. A business which showed a good profit in 1919, at the height of the boom, may have shown none in 1921, at the depth of the depression; and yet it may have called for more skill to have avoided a receivership in 1921 than to have earned the dividend in 1919. Any true measure must confine itself to those conditions over which management has control.

3 The other answer is, "No. Skill in management is too complex and too personal to be capable of measurement at all." This is the other extreme. Management must be, and is being, daily evaluated in *some* way, if not by measurement, then by personal judgment. The complexity and personal element are granted. Management, from this very complexity, is partly mechanistic, conforming to natural and sociological laws, and only in part

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personal. The terms "Science of Management" and "Art of Management" are both in general use. In so far as management is a science, measurement can and should be used, to the great advantage shown in every other science. Those elements which constitute an art, and are based on personal skill, leadership, and imagination, as well as the vital element of integrity, can and should be left to judgment as heretofore. Where applicable, measurement is more trustworthy. It should therefore be applied wherever possible.

4 *What Does Management Cover?* Management is recognized to cover a much wider field than formerly. This is reflected in the transactions of the A.S.M.E., which contain the most complete record of its development. The earlier volumes deal only with the technical aspects of production. In 1886 Mr. Towne proposed that the Society undertake to gather information on industrial management. In 1889 he gave his paper on Gain Sharing, and Mr. Halsey's paper on the Premium Plan was given in 1891. In later volumes one sees the range of topics constantly widening. Mr. Taylor's work was, at first, also directed toward finding a satisfactory basis for wage payment, but his investigation steadily broadened until it included a whole system of shop management, cost accounting, routing and scheduling, etc. Today management is generally recognized to include not only production, but finance, purchasing, sales, accounting, and personnel administration.

5 The methods of measurement proposed in this paper will be confined chiefly to the field of production, although it should be recognized from the start that production is only one of a number of equally important functions. This limitation is made because it seems at present to be the field in which measurement can be most readily applied and with the greatest promise of success. The development of a workable method covering production may make it easier to develop methods for other fields.

6 *The Need of Measurement.* The management function has been *judged* since the days of the stone age, when men first began to work collectively. Until recently, however, there has been little thought of actually *measuring* it.

7 There is a great difference between judgment and measurement. To judge is to arrive at a conclusion or a decision by weighing or comparison. To measure is to ascertain quality, dimensions, or quantity by comparison *with a standard.* In judging things, they are compared one with another. In measuring them, all are compared with the same thing, namely, the standard. Comparison is common to both, but the use of an agreed-upon standard as the basis for comparison distinguishes measurement from judgment. Judgment is the more widely applicable process. Measurement is possible only where a standard can be used. Measurement, where applicable, is the more definite and satisfactory, because all agree to compare with the same thing, the standard.

8 If practicable standards can be set up for at least some of

the managerial functions and suitable methods of comparison also agreed upon, a part at least of the personal element used in evaluating management can be eliminated; and in so far as measurement can be thus substituted for judgment, to that extent will the rating become more trustworthy.

9 There is increasing need for more trustworthy evaluations of management because of:

a The increasing scale of production. Mistakes in management are more disastrous now than when production was on a smaller scale

b The widening of competition to a national and even international basis. To compete with foreign labor American productive facilities must be used to maximum advantage

c The narrowing margin of profit. In the easy-going days a generation or two ago, profits were large enough to permit inefficiencies of management which today would be fatal

d The development of management as a separate function. Formerly the manager was usually the owner. Now, more and more, he is acting for others who are the owners, and is responsible to them. It is desirable that the owners have better means for gaging the effectiveness with which their business is being managed, and that the managers be able to prove their work to the owners

e The diversity of the points of view from which management is, and must be, judged. The interests and personal views of those making these judgments lead to different estimates for the same set of conditions. In so far as they can agree upon a common method, it will be a benefit to all

f The rapid development in management methods. New methods are being tried, some wise, some not. We should be able to determine their effectiveness quantitatively, if possible; to pass judgment on them on the basis of actual fact instead of predisposition, enthusiasm, or habit

g It is desirable to have some form of progressive judgment, to know what the management *is accomplishing*, not what it has done after crucial mistakes have been made.

10 It is a healthy sign that responsible industrial engineers and executives are the very ones who are most desirous to have their work measured. They are calling for it and will welcome it as a means for assisting them in their work.

11 *The Effect of Measurement in Other Sciences.* The great development of modern science has been based upon improvements in measurement. Before the application of measurement the sciences were largely speculative, practically useless, sometimes dangerous, and often a cloak for imposture. With the develop-

ment of measurement they have grown exact, effective, and useful. Natural philosophy has become physics, alchemy has become chemistry, astrology has become astronomy, and modern medicine and surgery have developed from the blood letter and chirurgeon of the Middle Ages.[1]

12 There is every reason to believe that in so far as measurement can be applied to management, it will be of similar benefit. Measurement in management should

a Increase its practical utility

b Permit action with more practicable results, and

c Tend to sift the wheat from the chaff both in men and methods.

13 *Use of Proven Methods.* The answer to a new problem often comes through the application of methods and means which have been in successful use in other fields. We now have satisfactory means for measuring the performance of machines and of workmen in connection with commercial production, which are in daily use and of incalculable value in modern production. It is well to consider how far these may be applicable in the field of management.

Measurement in Engineering

14 Three elements are characteristic of measurement as applied to commercial production:

a An agreed-upon standard, the qualities of which we seek to approach or duplicate

b Agreed-upon units of measurement applicable both to the standard and to the things to be measured

c Means or methods of *impersonal* comparison of the thing to be measured with the standard by means of the agreed-upon units. Impersonality is the soul of measurement and is one of the elements which differentiate it from judgment. Different persons, properly qualified and in possession of the same facts, should be able substantially to duplicate the results.

15 As an embodiment of these three characteristics, we might have

As the standard (*a*), the barrel of a model rifle

As units of measurement (*b*), inches, degrees of curvature, hardness, etc. These can be used to express the desired qualities so completely that 10,000 barrels can be made from well-dimensioned drawings without the use of the model barrel

As methods of impersonal comparison (*c*), plug and ring gages, contour gages, etc., whereby any properly qualified person can determine the conformity of the production barrels to the model within the limits desired.

[1] Medicine is the least exact and certain of the sciences mentioned just because it is the one in which measurement is most difficult.

16 *Phases Affecting the Problem.* The use of these elements in actual production is scientifically correct and perfectly practical. They are used in the production of millions of dollars' worth of material every day. It is worth while to see if their use cannot be extended from the material side of production to the more intangible executive side. In doing so, however, three phases should be borne in mind:

17 (*a*) *Break-Up into Functions.* It is a fundamental principle in science and engineering that a problem involving several variables can be solved only by segregating each variable, solving it separately, and then combining the results. Babbage recognized this as applying with equal force to problems of manufacture in his remarkable "Economy of Manufactures," written nearly 100 years ago. Management is a complex function involving a large number of variables, and no satisfactory rating for it can be obtained unless this procedure is followed. This breaking down into separate elements was the basis upon which Mr. Taylor built up his methods of time study. The only hope for measuring management successfully is to extend his general method from the study of a single task to that of management as a whole.

18 (*b*) *Measurement by Reactions.* Science and engineering continually deal with quantities not directly measurable upon the principle that action and reaction are equal. If we can identify a cause and its effect, and can measure that effect, we have a measure of the cause of that effect. This principle may be utilized in this problem in evaluating some of the executive elements which are not themselves directly measurable.

19 (*c*) *Extreme Accuracy Not Essential.* Measurement does not have to be extremely accurate to be useful. If we measure in the same way each time, with the same units, and recognize that the results are approximate, we are far better off than with no measure at all.

Characteristics Necessary in Any Method of Measuring Management

20 Any method of measuring management which has any hope of being successful must have the following characteristics, some of which have already been referred to:

a It must be sound, clear, and convincing

b It must be based on ascertainable facts

c It must reflect the conditions under which the management operates

d It must attempt to measure only such elements as are measurable

e It must be confined to those elements over which the management has control

f It must eliminate, as far as possible, the element of opinion in application, that is, be impersonal

g It must offer a better and sounder basis for measuring the managerial function than we now have

h It must be useful. It should give more than an overall efficiency, and should indicate, if possible, the relative strength and weakness of the various elements to serve as a guide for development and improvement

i It must be as simple as is consistent with the complexity of the problem to be solved.

The author feels that the various methods suggested up to this time fail in one or more of the characteristics given above. The general procedure used in scientific and engineering measurement can be adapted to the measurement of some, at least, of the elements of management in a way which will satisfy these conditions.

21 With the above considerations in mind the following method is suggested:

a Break down the managerial function into component elements which may be considered separately

b Set up an ideal performance for each element

c Set up a standard of performance for each element which shall, if possible, conform to this ideal; if this is not possible, one which will give a practical working basis approaching that ideal and will mean the same thing to all. If *no* standard seems practicable at this time, this element may be omitted from the rating

d Set up a method of rating the performance of the management against each of these standards

e Develop a method of bringing together the individual element ratings, properly weighted, into a composite rating.

22 The final rating should be used with the understanding that it covers only those elements at present considered measurable and that the remaining elements, if any, which do not lend themselves to measurement are still to be evaluated on the basis of judgment.

23 The application of the method which follows is given as illustrating a method of procedure. A more satisfactory division into elements is quite possible. The same is true for the standards and rating ratios set up. These can be established best by those familiar with the conditions in each industry. It is the method only which is emphasized. It will be confined to the field of production, because it is difficult at this time to set up satisfactory standards of performance and methods of ratings in the other fields. Progress is being made in this direction, however, and it is possible that standards and methods for measuring these elements will be developed. The method of analyzing financial statements given in the pamphlet issued by the Robert Morris Associates [1]

[1] "Financial Statements" — An explanation in brief of a system for analysis from the standpoint of the credit grantor and the business executive. Issued by the Robert Morris Associates, Lansdowne, Pa.

is a step in this direction. The general financial statement is analyzed into eight ratios. These are weighted in accordance with the conditions of that industry, and brought together into a single rating based on unity. This approaches very closely the procedure set forth in this paper.

24 *Division into Elements.* For our present purposes production will be divided into the following elements:

- *a* Purchasing
- *b* Stores Inventory
- *c* Efficiency of Plant, Equipment, and Methods
- *d* Efficiency with Respect to Materials Used
- *e* Efficiency with Respect to Payroll
- *f* Idleness of Equipment
- *g* Quality of Output
- *h* Keeping of Promises
- *i* Labor Turnover
- *j* Cost Accounting and Records.

With each of these elements we will consider first the ideal performance in respect to this element; second, a practical standard for it, if one is possible; third, a rating ratio for measuring the performance against this standard.

Purchasing

25 *Ideal.*

- *a* The securing of the right amount of material, of the right kind and quality
- *b* At the lowest prices available, and
- *c* Having this material on hand when it is wanted, where it is wanted, and in the condition wanted.

26 *Practical Standard.* The "right amount" is the least which will provide an uninterrupted supply for production and will maintain the minimum stores required for safe operation. If the proper minimum of the stores of the right kind and in the right condition is maintained at all times, there should be no interruptions of production from failure of raw-material supply. Items (*a*) and (*c*) above will be cared for under the element of Stores Inventory.

27 The "lowest prices *available*," item (*b*), are not readily ascertainable, and with long-time purchases are complicated by the cost of carrying charges. Current market prices are, however, usually obtainable and may be used in setting up a practicable working standard. Furthermore, in most manufacturing industries the bulk of the value of the total purchases is concentrated in a few basic raw materials.

28 If we could set up a ratio between the actual cost of these few basic materials used each day and their cost if bought "over the counter" at the market price that morning, we would have a fairly good measure of the purchasing efficiency for these materials. If

the actual purchase cost were less than the current market, it would, when divided into the market cost, show a purchasing efficiency above par. If it were the same, the purchasing efficiency would be at par. If the purchase cost is greater, the ratio will be less than unity and the purchasing is below par. If this ratio is found to be favorable for the basic materials, the multitudinous minor supplies which make up the relatively small balance may be disregarded.

29 *Rating Ratio.* It would be impractical to make this comparison on a daily basis, and comparison with the monthly average market prices would, in most cases, be sufficient. A practical rating for the purchasing function therefore may be expressed as follows:

$$\text{Rating ratio} = \frac{M_1 + M_2 + M_3, \text{ etc.}}{C_1 + C_2 + C_3, \text{ etc.}}$$

where

M_1, M_2, etc., are the products of the monthly consumption of a selected basic material multiplied by the *average market price* for that material during that month, and
C_1, C_2, etc., are the actual *costs* of these materials *as purchased.*

30 As pointed out above, if the materials have been purchased at low prices the values in the denominator will be less than the corresponding ones in the numerator. If desired, the ratio for each material may be taken separately, these added, and the sum divided by the number of materials to bring the result to a unit basis. Where the material has been carried a long time, interest and carrying charges may be added to the purchase cost, if that refinement is deemed necessary. Both numerator and denominator are in dollars and the quotient is a ratio for that month which may be less than, equal to, or greater than unity, according as the given materials have been bought above, at, or below the average market prices for that month.

31 The rating of the purchasing element may be based on the running average of the ratios for the past twelve months instead of a single monthly ratio. Such an average would be fairer than a single ratio, as it will show how well the purchasing department is anticipating prices.

Stores Inventory

32 *Ideal.* The maintenance at all times of the lowest amount consistent with having all the material ready for process, when, where, and in the condition needed.

33 *Practical Standard.* For simplification, instead of considering all the stores, the same basic raw materials may be used for this element as in the case of purchasing.

34 Set up for each basic material the minimum safe time which should be allowed to order and obtain that material, with due consideration to distance, reliability of source, contingencies, etc., in accordance with the present well-established practice. These times may, and probably will be, different for each material, and may vary with market conditions.

35 If three months, for instance, was the safe time allowance on, say, January 1 for obtaining one of these materials, then the sum of the consumptions during January, February, and March is the amount of stores which should have been on hand at that date. This can be used as a standard, and can be divided into the actual stores of the material on hand at that time. This will give a ratio which will be less than, equal to, or greater than unity according as the stores actually on hand were less than, equal to, or more than the safe amount needed. A similar ratio may be set up for each of the basic materials. A moving average for, say, one year of the deviation of the percentages from unity for each material, *disregarding the sign*, that is, whether the deviation of the percentages is above or below unity,[1] will indicate how close the stores inventory on that material had been kept to the desirable minimum.

36 To obtain the ratio for the stores inventory as it stands at the *present* time it is necessary to use the estimated requirements, through the production or sales schedules, as this is the best information we have.

37 *Rating Ratio.* The rating ratio can be obtained from the average of the moving averages for the various materials, again disregarding the sign, i.e., whether the deviation is above or below unity. This average deviation should always be *deducted* from unity to obtain the rating.

38 Unity in this rating would indicate that the management had uniformly maintained the stores inventories of its basic materials at the minimum which had been established as good practice and under the conditions of operation. Ratings less than unity would indicate how far the inventory had been departing from this standard. Examination of the monthly figures would show when, in which materials, and in which directions the deviations had occurred.

39 For the detail planning and follow-up of the material stores we have in the Gantt layout and progress charts a useful and tried-out method which in itself is an excellent measure of the performance of this function. It does not, however, head up into a single rating as the method suggested does, but can readily be used in connection with it.

[1] If the signs of the deviation are not disregarded, plus and minus deviations would offset each other and tend to show conformity to the required quantities when such conformity did not exist. If the deviations are averaged arithmetically without regard to whether they are above or below, we will get a true measure of the conformity.

Effectiveness of Production

40 *Ideal.* The production of the greatest value of salable output with the least expenditure for plant, materials, and labor.[1] This has three aspects: production efficiency with respect to

a Plant, equipment, and methods
b Materials
c Labor employed.

41 A management may be making poor use of a good plant and equipment, or the best possible use of a poor equipment. It is desirable, therefore, to measure these phases separately. This can be done roughly by dividing the product sold per annum in case (*a*) by the plant investment, in case (*b*) by the materials purchased annually, and in case (*c*) by the total annual payroll. The products *sold* is in general a better value to use than the value of the goods produced, because it represents goods which actually contributed to the income. In certain businesses, such as heavy machinery or shipbuilding, however, the goods produced, including even those only partly finished, may be the better value to use.

42 *Practical Standard.* These ratios will of course vary with different industries, and it is necessary to compare them with some standard before we can know whether they are high or low. The most practical standard seems to be the similar ratio for the industry as a whole. Various Government reports such as the 1919 Census of Manufactures and the Statistical Abstract of the U.S. give the capital invested, value of the products, value of the materials used, and total payroll for many industries.[2] There are other sources also, such as trade associations, so that in most cases the information necessary for the comparison is available.

43 *Rating Ratios.* If the ratios for the industry are available the rating for efficiency with respect to plant, equipment, and methods would be

$$\frac{\text{Annual sales in dollars}}{\text{Plant investment in dollars}} \div \frac{\text{Total annual output of the industry}}{\text{Total investment in the industry}}.$$

The result will be above, equal to, or less than unity according as the ratio for the firm in question is better, equal to, or worse than the average of its competitors. In times of extreme depression it may be fairer to divide this ratio by the idleness ratio (Par. 49).

44 The rating with respect to materials would be

$$\frac{\text{Annual sales in dollars}}{\text{Total annual purchase of materials}} \div \frac{\begin{array}{c}\text{Total annual output}\\ \text{of the industry}\end{array}}{\begin{array}{c}\text{Value of raw materials used}\\ \text{in industry per annum}\end{array}}.$$

[1] With respect to labor this should be construed as the least expenditure consistent with sound social conditions, to exclude child labor, etc.

[2] Appendix No. 1 gives these data for certain selected industries. The Statistical Abstract of the U.S. gives them for more than 300 industries, in addition to those listed in the appendix.

Similarly, the rating with respect to use of labor would be

$$\frac{\text{Annual sales in dollars}}{\text{Total annual payroll in dollars}} \div \frac{\text{Total annual output of the industry}}{\text{Total annual payroll of the industry}}.$$

These ratios may be above unity and will show ratings on the basis of the average for that industry or the average of the competition which the firm has to meet.

45 The material turnover would also be a useful element to measure, but no standards or methods of rating seem to be available. If these could be satisfactorily set up it would be desirable to include it.

Idleness of Equipment

46 The degree of approach of the actual output to the full capacity of the plant is a significant means for measuring management. In reality it is included in the plant-efficiency ratio above, but it shows directly what use is being made of the equipment and is so easily obtained and is so useful that it is well to set it up separately. Mr. Gantt told the author that he used it continually in diagnosing the situation in a plant.

47 *Ideal.* The ideal would of course be to have all the equipment at work at full capacity all the time. Of course, also, this is impossible.

48 *Standard.* The same, i.e., 100 per cent or the full capacity of the plant.

49 *Rating Ratio.* This would be

$$\frac{\text{Actual production machine hours}}{\text{Machine hours' capacity of the plant}}.$$

For each industry the ratio would have a characteristic maximum, always less than unity.

Quality

50 *Ideal.* That all goods produced pass all inspections, both producer's and customer's.

51 *Standard.* The same, i.e., 100 per cent of the goods produced.

52 *Rating Ratio.* This would be

$$\frac{\text{Goods produced which pass all inspections}}{\text{Total goods produced}}.$$

This rating will presumably always be less than unity, and it may be in certain cases, as in some garment industries, that a moderate proportion of "seconds" is preferable to the expense of trying for perfection.

Promises

53 *Ideal.* All promises kept.

54 *Standard.* The same.

55 *Rating Ratio.*

$$\text{Rating ratio} = \frac{\text{Number of promises kept}}{\text{Number of promises made}}.$$

This rating also will, in general, be less than unity.

Labor Turnover

56 *Ideal.* That turnover, only, which is due to inevitable causes, such as death, superannuation, marriage; seldom, if ever, attainable.

57 *Standard.* The turnover prevailing in that trade and locality, so far as these can be determined.

58 *Rating Ratio.*

$$\text{Rating ratio} = \frac{\text{Turnover of plant under consideration}}{\text{Turnover of that trade and locality}}.$$

Cost Accounting and Records

59 *Ideal.* The gathering and maintaining of information on all costs which shall be

a Accurate
b Prompt
c Useful, *and used*
d Tied in with the production and financial accounts

60 This element is vital in modern management. The author has been unable to discover a standard or means of *measuring* performance in it which would be satisfactory or generally applicable. The best way would be to recognize this frankly; to set up an ideal and rate the performance of the management in this element on the basis of judgment of men properly qualified to pass on it.

The Final Rating

61 The element ratings may be brought together into a final rating as follows:

	(1) Element	(2) Individual rating of element	(3) Percentage weight of element in total rating	(4) Weighted rating of element (col. 2 × col. 3)
a	Purchasing	....	....	....
b	Stores Inventory	....	....	....
c	Efficiency of Plant, Equipment, and Methods	....	....	....
d	Efficiency with Respect to Materials Used	....	....	....
e	Efficiency with Respect to Payroll	....	....	....
f	Idleness of Equipment	....	....	....
g	Quality of Output	....	....	....
h	Keeping of Promises	....	....	....
i	Labor Turnover	....	....	....
j	Cost Accounting and Records (on basis of judgment)	....	....	....
			100%	
	Final Rating	....	....	....

62 The separate elements and their ratings as obtained above may be listed as in columns 1 and 2. Percentage weights for each element, suitable for that industry, are set up in column 3; and the weighted value for each element (column 2 $\times$ column 3) is extended in column 4. The final or combined rating will be the sum of the items in column 4. Values above unity in the element ratings for *a*, *c*, *d*, *e*, and *i* would indicate that the quality of the management, in these elements, was better than the average of the industry; values below, that it was poorer.

63 The ratings in column 2 are founded on ascertainable facts and are largely free from opinion or judgment. The weights in column 3 assigned to the various elements, however, must involve judgment, which will of course affect the weighted ratings in column 4. If this is objected to, the use of the method can stop with column 2. A periodic comparison of the variations in the element ratings in that column will, alone, have value in guiding management policies. If our purpose is to measure the management as a whole we must recognize that the elements will have different relative importance in different industries, and they must be weighted to reflect the conditions for each industry. After all, as we have said, extreme accuracy is not required. If some weighting can be *agreed upon* for the industry in question, and all concerned will use that weighting, then column 4 and the Final Rating will have value as a basis for comparison.

Utility of the Above Method

64 The method outlined furnishes a basis for comparing managerial performance in different plants of the same or allied industries, but more particularly it offers a means of following variations in the management of a single plant.

65 The element ratings and the final rating may or may not have considerable significance where applied once to a specific business, as of a certain date. If, however, the method be used repeatedly, in the same way, each successive set of ratings will become more trustworthy and useful because we have set up a uniform and orderly basis of comparison for the measurable elements of the executive function.

APPENDIX NO. 1

INFORMATION ON CERTAIN INDUSTRIES FOR THE CALENDAR YEAR 1919

Note. — The industries given below are selected out of more than 350 listed in Table No. 160, p. 226, of the Statistical Abstract of the U.S., 1921, from which columns 1 to 5 are taken. The ratios given should be used with caution as the year 1919 was one of abnormal production. The table brings out clearly, however, how widely the ratios vary as between different industries.

1	2	3	4	5	6	7	8
					Ratios of Value of Annual Output to		
Industry	Capital, Dollars	Wages, Dollars	Cost of materials, Dollars	Value of products, Dollars	Capital	Wages	Cost of Matls.
Agricultural implements	366,962,052	66,704,434	144,571,943	304,961,265	0.83	4.57	2.11
Automobile bodies and parts	470,497,552	178,955,503	362,027,302	692,170,692	1.47	3.87	1.91
Automobiles	1,310,451,400	312,165,870	1,578,651,574	2,387,903,287	1.82	7.65	1.51
Boots and shoes, not including rubber boots and shoes	612,625,075	210,734,610	715,269,315	1,155,041,436	1.88	5.48	1.62
Boots and shoes, rubber	131,513,436	30,882,722	50,346,880	116,917,434	0.89	3.78	2.32
Brass, bronze, and copper products	325,299,738	94,132,118	304,823,580	482,312,790	1.48	5.13	1.58
Bread and other bakery products	529,265,779	158,237,059	713,239,411	1,151,896,318	2.18	7.29	1.62
Brick and tile, terra cotta, and fireclay products	355,848,355	78,256,085	67,488,113	208,422,920	0.59	2.67	3.10
Butter	162,302,108	19,052,729	514,345,739	583,161,011	3.60	30.60	1.13
Canning and preserving, fruits and vegetables	223,692,234	43,592,537	265,628,525	402,242,972	1.80	9.23	1.52
Cars and general shop construction and repairs by steam-railroad companies	694,286,410	687,617,312	515,803,210	1,279,235,393	1.84	1.86	2.48
Cars, steam-railroad, not including operations of railroad companies	335,207,363	78,284,647	356,084,545	538,222,831	1.61	6.88	1.52
Cement	271,269,259	33,194,920	79,509,800	175,264,910	0.65	5.29	2.21
Chemicals	484,488,412	72,848,324	216,301,279	438,658,869	0.91	6.02	2.03
Clothing, men's	554,147,279	197,821,990	605,752,176	1,162,985,633	2.10	5.90	1.92
Clothing, women's	390,526,517	195,295,834	680,406,844	1,208,543,128	3.09	6.19	1.77
Coke, not including gas-house coke	365,249,622	42,299,292	224,266,673	316,515,838	0.87	7.49	1.41
Confectionery and ice cream	317,043,923	76,159,866	368,809,170	637,209,168	2.01	8.37	1.72
Cotton goods	1,853,099,816	355,474,937	1,277,785,597	2,125,272,193	1.15	5.98	1.66
Dyeing and finishing textiles, exclusive of that done in textile mills	299,948,486	57,189,978	174,742,815	323,967,683	1.41	5.66	1.85
Electrical machinery, apparatus, and supplies	857,855,496	238,188,852	425,098,211	997,968,119	1.16	4.20	2.34
Engines, steam, gas, and water	454,124,733	105,435,455	217,550,771	464,774,735	1.02	4.41	2.14
Explosives	133,247,684	12,504,986	45,911,049	92,474,813	0.69	7.40	2.01
Flour-mill and grist-mill products	801,624,507	50,888,383	1,799,180,987	2,052,434,385	2.56	40.40	1.14
Food preparations, not elsewhere specified	245,282,687	29,392,209	494,597,157	631,598,150	2.58	21.50	1.28
Foundry and machine-shop products	2,104,980,938	622,571,129	948,069,381	2,289,250,859	1.09	3.68	2.41

Furniture	423,992,405	141,116,316	261,523,395	571,376,333	1.35	4.05	2.18
Gas, illuminating and heating	1,465,656,265	52,758,628	157,5[illegible]0,882	329,278,908	0.22	6.24	2.09
Hardware	133,925,619	45,229,950	58,533,769	154,524,888	1.15	3.42	2.64
Ice, manufactured	270,725,786	34,001,837	42,877,509	137,004,798	0.51	4.03	3.20
Iron and steel, blast furnaces	802,416,541	73,769,395	621,286,496	794,466,558	0.99	10.80	1.28
Iron and steel, steel works and rolling mills	2,656,518,417	637,637,430	1,680,575,758	2,828,902,876	1.06	4.44	1.68
Knit goods	516,457,991	125,199,820	427,095,560	713,139,689	1.38	5.69	1.67
Leather, tanned, curried, and finished	671,341,553	88,205,473	646,521,527	928,591,701	1.38	10.51	1.44
Lumber and timber products	1,357,991,571	489,419,091	470,960,488	1,387,471,413	1.02	2.84	2.95
Lumber, planing-mill products, not including planing mills connected with sawmills	361,848,079	91,976,526	299,265,652	500,438,[illegible]58	1.38	5.45	1.67
Machine tools	231,039,843	66,178,969	59,034,308	212,400,158	0.92	3.21	3.59
Oil and cake, cottonseed	203,457,371	20,615,193	495,192,294	581,244,798	2.86	28.20	1.17
Paints	177,314,815	19,550,371	165,604,116	256,714,379	1.45	13.12	1.55
Paper and wood pulp	905,794,583	135,690,642	467,482,637	788,059,377	0.87	5.82	1.69
Petroleum refining	1,170,278,189	89,749,637	1,247,908,355	1,632,532,766	1.39	18.20	1.31
Printing and publishing, book and job	446,554,984	141,476,243	211,067,174	597,663,228	1.34	4.23	2.83
Printing and publishing, newspapers and periodicals	614,045,344	144,348,173	300,385,187	924,152,878	1.51	6.40	3.15
Rubber tires, tubes, and rubber goods, not elsewhere specified	782,637,722	156,806,828	525,686,309	987,088,045	1.26	6.29	1.87
Shipbuilding, steel	1,268,640,254	538,372,576	643,752,814	1,456,489,516	1.15	2.70	2.26
Shirts	102,012,047	25,833,855	127,087,745	205,327,133	2.01	7.95	1.62
Silk goods	532,732,163	108,226,330	388,469,022	688,469,523	1.29	6.36	1.77
Slaughtering and meat packing	1,176,483,643	209,489,263	3,782,929,533	4,246,290,614	3.61	20.30	1.12
Smelting and refining, copper	308,680,268	25,723,371	584,410,173	651,101,591	2.11	25.30	1.12
Soap	212,416,866	21,228,063	238,518,858	316,740,115	1.49	14.90	1.33
Tobacco, cigars, and cigarettes	416,395,472	111,313,348	353,297,366	773,662,495	1.86	6.95	2.19
Tools, not elsewhere specified	134,731,947	43,836,069	45,796,967	144,201,668	1.07	3.29	3.15
Steam fitting and steam and hot-water heating apparatus	133,097,464	45,742,525	72,016,393	160,285,488	1.20	3.50	2.22
Stoves and hot-air furnaces	122,813,373	41,321,133	54,803,316	145,717,963	1.19	3.53	2.66
Structural ironwork, not made in steelworks or rolling mills	219,470,095	59,920,132	168,800,715	294,962,419	1.34	4.92	1.75
Sugar, beet	224,584,679	15,908,118	87,029,144	149,155,892	0.67	9.40	1.71
Sugar refining, not including beet sugar	193,540,825	22,710,464	662,143,981	730,986,706	3.79	32.19	1.11
Wire	102,016,777	29,289,667	102,813,591	162,151,236	1.59	5.55	1.58
Woolen and worsted goods	831,694,748	168,108,681	665,594,683	1,065,434,072	1.28	6.34	1.60

APPENDIX NO. 2

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No. 1915

PRINCIPLES OF LONG-DISTANCE AIR NAVIGATION

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This paper contains a general account of the scientific principles of long-distance air navigation, including dead reckoning, astronomical observations, and radio methods. It has been prepared at the invitation of the Aeronautic Division and is addressed to those who may be interested in a broad way in the problems and possibilities of commercial aeronautics. Hence it has not been thought necessary to enter into detail except where the methods presented are believed to be new. As a guide to those who will wish to pursue the subject farther, the paper concludes with a fairly complete outline of the most recent sources of technical information.

I—INTRODUCTION

THIS PAPER deals with the general principles of long-distance air navigation, including instruments and methods required for determining the position of an aircraft when the ground is not visible. It has been prepared as one of a group of papers treating different phases of commercial aeronautics, and is intended for the general reader rather than for those professionally engaged in the solution of navigation problems.

2 Following a general survey of the problems and difficulties by which air navigation is differentiated from marine navigation, a more detailed account is given of the three fundamental systems of navigation: dead reckoning, astronomical navigation, and radio navigation. In this connection (Sections IV, V, and VI, respectively) reference is made to the various methods and instruments

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in recognized use, and in several instances new methods are proposed.

3 Various expedients for safety and efficiency are summarized in Section VII, and the troublesome problems of cloud flying and fog landing are here briefly discussed.

4 Recent instrument developments, as well as research problems awaiting future attention, are outlined in Section VIII. Section IX is a résumé of the main sources of technical literature, together with a summary of advances actually made in air navigation during the past few years.

II—GENERAL VIEW OF THE DIFFICULTIES INVOLVED

5 Air navigation evidently differs from marine navigation in the following advantageous respects: (1) The craft can usually ascend above the clouds for the purpose of astronomical observations; (2) drift determinations can frequently be made by sighting directly on the ground or water passed over, without previous data on wind currents; and (3) an error of, say, 20 miles (15 minutes flying time) in the determination of absolute position is generally negligible. On the other hand, air navigation has the following characteristic disadvantages: (1) The impossibility of the craft standing still for any purpose; (2) a much greater velocity of drift; (3) the impracticability of charting the atmospheric currents with precision; (4) the difficulty of holding the ship steady for astronomical or other observations; (5) the complete absence, at times, of any reliable natural horizon; (6) the complexity of dealing with three dimensions instead of two; (7) the necessity for correlated observations to be taken practically simultaneously; and (8) the importance of carrying through computations of position with great despatch, in order that course corrections can be executed before losing ground.

6 These disadvantages of air navigation, compared to marine navigation, so far outweigh its advantages that the problem of air navigation is a difficult one. It may be considered solved at the present date in nearly every respect; but to accomplish this, all resources of meteorological knowledge, good piloting, scientific-instrument development, skillful observation, and short-cut computation methods are required, and there will still be room for improvement through future research. The only respect in which air-navigation difficulties have not been surmounted appears to be the case of forced landing in fog when flying over unfamiliar territory, which will be considered in Section VII.

III—FUNDAMENTAL METHODS OF AIR NAVIGATION

7 *Dead reckoning* consists in calculating the true course, distance, and final position of the aircraft solely from observations of speed or distance relative to the air, together with a knowledge of the ship's

compass heading from time to time, and data concerning the wind velocity.

8 *Astronomical navigation* involves the determination of position without reference to the foregoing data, purely in terms of astronomical observations and the time by chronometer. Such observations require the simultaneous determination of the altitudes of two celestial bodies, or the altitude of one such body and its azimuth.

9 *Radio navigation* is the determination of position by any combination of wireless signals, with or without the use of a compass. This may be done either by the use of a wireless direction finder installed on the craft, signals being received from fixed stations of known location; or simply by sending out a call from the ship, requesting fixed stations which are suitably equipped to compute the position of the ship from their own observations and to report it back by wireless message.

10 It is not necessary to adhere exclusively to any one of these three systems, therefore seven different combinations are possible, as well as numerous variations of procedure or equipment when applying any one of the fundamental methods. By no means have all the combinations possible been tested in practice as yet, or even theoretically analyzed.

11 Besides the three methods mentioned, two others are also possible, though of less interest for the present discussion; flying by map over visible ground is one such method, and at the other extreme, absolute position finding by gyroscopic methods.

12 There is a definite technique and art of map flying, involving the use of special aeronautical maps leading to ready recognition and identification of the territory viewed from the ship. This method may be facilitated by conspicuous markings on the ground below, as well as by convenient devices on board ship for holding and adjusting the maps.

13 Neither will the problem of navigation by purely gyroscopic means be further considered, as it is not known that any research on this subject has yet reached the stage of practical application under aeronautical conditions. That gyroscopic apparatus may, however, be employed for navigation as an auxiliary in conjunction with other instruments, is evidenced by the popularity and success of the turn indicator.

IV—DEAD RECKONING

14 The problem of determining the path of an airplane in the air resembles that of determining the path of a ship on the sea, which is affected by the action of ocean currents. But in general the velocity of ocean currents is so much less than that of the ship that the allowance for their effect is relatively small. A steamship having a speed of 15 knots will rarely be for any considerable time subjected to a current of 2 knots, while an airplane with a speed of 75 m.p.h. will not infrequently encounter winds of 30 m.p.h.

15 It is important, therefore, for the airplane to have some means of determining the direction and speed of its motion over the ground due to the resultant of its motion through the air compounded with the motion of the air over the ground.

16 This is now usually done by finding the "drift" by observing the direction of the apparent motion of some easily distinguishable object on the ground beneath, with reference to the line of the keel of the airplane. Such methods require direct observation of the ground and will be described below. Later, a method will be

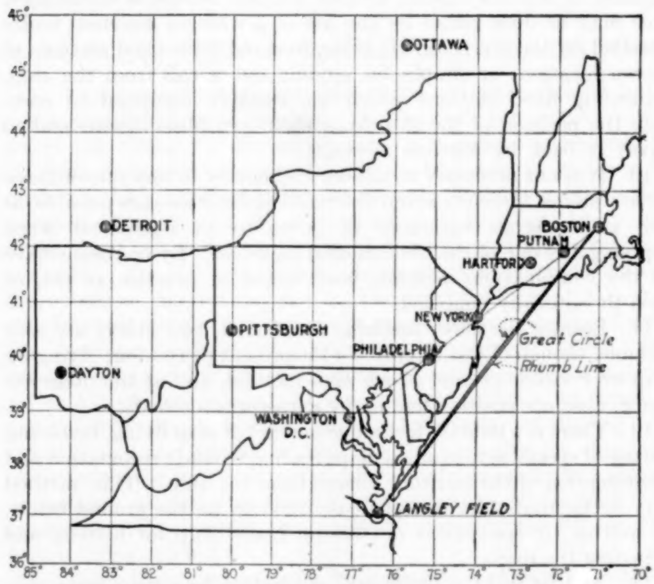

Fig. 1 Northeastern States, Mercator Chart

discussed which does not require ground observations and would therefore be available for dead reckoning above the clouds or in fog.

Elementary Principles

17 While the determination of position by dead reckoning theoretically requires continuous observations, it is customary to divide up the computation into successive intervals, during each of which the drift, compass course, and air speed may be considered constant.

18 To make the matter concrete, consider the particular case of an airplane starting out to fly from Langley Field to Boston (Fig. 1). Suppose the air speed is $A = 75$ m.p.h. and that the wind is blowing from the west with a speed $W = 30$ m.p.h. at the height of the airplane.

19 Examination of the map (Fig. 1) shows that the true course to be flown is approximately north 39 deg. east. Let it be required to determine the proper compass course for the pilot to steer.

20 In Fig. 2, the wind vector W is laid off from the origin O toward the east with magnitude 30 m.p.h., while a line drawn through the origin at the angle $g = 39$ deg. from the north represents the true course to be flown. Where shall the vector A, representing the air speed and therefore the direction of the axis or keel of the ship, be placed so that the resultant of A and W shall lie in the desired direction g?

21 A simple construction to locate A, having given the mag-

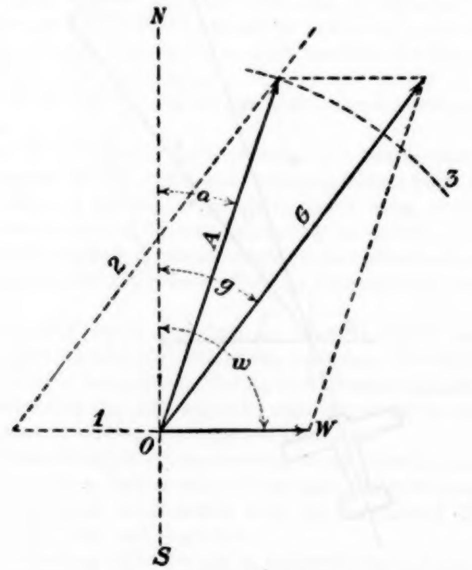

Fig. 2 True Course, Compass Course, and Wind Direction

nitudes of A and W and the angle g, is as follows: Reverse the vector W (line 1 on the diagram, Fig. 2); starting at the end of this line, draw another (line 2) of indefinite length parallel to G. Then with O as center, and radius A, draw a circular arc 3; the intersection of this arc with line 2 is the required location for the extremity of the air-speed vector A.

22 The pilot should now set the keel of the ship along the line A, and the compass course to be steered is shown on the diagram by the angle a. The magnitude of a, as well as ground speed G, can be scaled from the diagram, or computed trigonometrically.

23 A simpler method for determining such quantities will

be explained in a later paragraph in connection with the use of the graphic traverse table (Fig. 9). The solution is found to be approximately $a = 21$ deg. east of north, while the drift angle y, hereafter referred to, is about 18 deg.

24 The essential data of Fig. 2 are reproduced in Fig. 3 in order to call attention to the relative wind angle x, drift angle y, and the keel speed K—quantities to be made use of later. The angle x is measured from the keel direction to the wind direction, while the drift angle is that subtended between the axis, or keel, of the ship

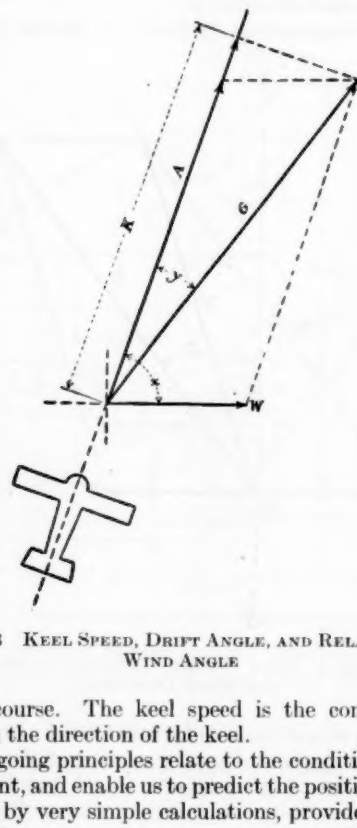

Fig. 3 Keel Speed, Drift Angle, and Relative Wind Angle

and her true course. The keel speed is the component of the ground speed in the direction of the keel.

25 The foregoing principles relate to the conditions of the flight at any one instant, and enable us to predict the position of the ship at some later time by very simple calculations, provided all quantities remain constant.

Instruments in Practical Use

26 Instruments for dead reckoning may be considered in two groups, namely, observing and computing instruments.

27 *Observing Instruments.* The actual observing instruments for dead reckoning are the compass, air-speed indicator, and stop

watch, together with various types of drift indicators, horizontal sights, and ground-speed indicators. Only the last three classes need be described here.

28 *Simple Drift Indicators.* Among the vertical sighting instruments for drift-angle observation may be mentioned the Sperry synchronized drift set, Webster drift sight, kinematic drift indicator (S.T.Aé.), Universal sight of the Compagnie Aérienne Française, and Pfadfinder drift compass.

29 The well-known Sperry drift set[1] consists of an observing telescope with cross-wires that can be set parallel with the "stream lines" caused by the apparent motion of the ground or water below. The drift angle may be read directly from a graduated circle. In the synchronized drift set the telescope control is connected with a compass by flexible wires so as to shift the lubber's line and enable the pilot to correct his course.

30 The S.T.Aé.[2] and Pfadfinder[3] sights are somewhat similar to the Sperry.

31 The D. L. Webster drift indicator is a simple wire fan or set of wires spaced 10 deg. apart and radiating from a point on the side of the fuselage at the rear edge of the lower wing, through which the apparent motion of objects below may be viewed.

32 The Compagnie Française sight[4] is an optical one installed in the floor of the cockpit, which offers an exceptionally wide field of view.

33 *Horizontal Sights.* Alidade or azimuth sights may be employed on aircraft with or without the compass. In the latter case they may be used for position finding by sighting on known objects, or for determining the drift angle by sighting ahead on some object which appears to maintain a constant azimuth.

34 Horizontal sights when attached concentrically to a magnetic compass serve in a similar way to indicate the true course of the ship. Among such instruments may be mentioned the Duval, Dunoyer, Bamberg, and Ludolph.

35 The Dunoyer sight[5] brings an image of the compass card into the line of vision by means of a prism. The Duval[6] instrument allows a graphical record of successive observations to be made. The Bamberg and Ludolph sights[7] differ only in mechanical details.

36 *Ground-Speed Indicators.* Possibly the earliest ground-speed indicator other than a simple bomb sight was the Sperry stroboscopic device.[8] Rotating telescopes serve to maintain the image of the ground in a position momentarily stationary so that a

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[1] E. A. Sperry, Trans. Soc. Auto. Engrs., vol. 1, pp. 158–161, 1917.

[2] A. B. Duval and L. Hébrard, Traité Pratique de Navig. Aérienne, pp. 42–43, 1922.

[3] R. Bennewitz, Flugzeuginstrumente, p. 188, 1922.

[4] Premier Cong. Int. de la Navig. Aérienne, Rapports, 3, p. 507, 1922.

[5] Duval and Hébrard, p. 45.

[6] Duval and Hébrard, p. 46.

[7] Bennewitz, pp. 192–193.

[8] Sperry, pp. 155–162.

knowledge of altitude and rate of rotation serves to fix the ground speed.

37 The Aero bearing plate[1] is a simple mechanical sight with graduated circle, showing the drift angle by vertical observations and ground speed by reference to stop watch and altimeter.

38 The Wimperis wind-gage bearing plate[2] is employed for observations off the tail of the ship. This instrument is a further development of the bearing plate mentioned above, such that wind speed as well as ground speed may be determined in magnitude and direction. These results may be secured while flying on a constant course by means of a stop watch; or by flying on two or more different courses, without a watch.

39 The Wimperis course-setting sight[3] is employed for sights taken ahead of the ship. It is a highly developed mechanical instrument of which a magnetic compass may or may not form an integral part. By flying on two different courses (up-wind and across-wind) the ground velocity and wind velocity can both be determined in magnitude and direction and read directly from the instrument, together with other information useful to the pilot. No stop watch is required.

40 The Crocco drift and ground-speed indicator[4] in conjunction with a stop watch serves to determine all the elements of the flight in a convenient automatic manner and is particularly adapted for dirigibles. It is mounted aft so that observations can be taken astern.

41 A simplified modification of the Crocco type has been developed by the Pioneer Instrument Company.[5]

42 Another calculating device mechanically connected to the observing element is the Maffert course corrector.[6] In place of the usual stop watch, a chronograph is employed, and the various elements of the flight can be read off directly.

43 A convenient mechanical device for determining drift, ground speed, and wind vector from observations on two courses was developed by Couthino and Saccadura,[7] whose names will be recognized in connection with their successful flight from Portugal to Brazil. The drift observations are taken on either side of the ship by reference to a system of lines drawn on the front edge of each wing and radiating from two points well forward on the fuselage.

44 The Cayère-Montagne[8] drift sight is of light construction and serves both for observing the drift angle and solving the velocity triangle.

[1] H. E. Wimperis, Air Navigation, pp. 48–50, 1920.
[2] Wimperis, pp. 53–55.
[3] Wimperis, pp. 55–58.
[4] H. N. Eaton, N.A.C.A. Rep. No. 131, pp. 17–19.
[5] Eaton, p. 19.
[6] Prem. Cong. Int., vol. 2, pp. 73–78.
[7] Prem. Cong. Int., vol. 2, pp. 112–114.
[8] Duval and Hébrard, p. 44.

45 In the Le Prieur navigraph,[1] drift observations are taken through a vertical telescope from time to time and plotted automatically on a sheet of paper by a pencil connected to the telescope. In this way irregularities due to unsteadiness of the craft are averaged out. By flying on two courses, the wind vector and ground speed can be determined and the course correction found, and this correction is transmitted to a repeating dial in the pilot's cockpit somewhat after the fashion of the Sperry drift set.

46 *Computing Devices.* Computers for dead reckoning include map protractors and velocity-triangle computers.

47 *Map Protractors.* Protractor developments for use on the chart board are well represented by the Bigsworth, Christmas, and Wichmann types.

48 The Bigsworth protractor,[2] by means of its parallel motion, permits plotting a given compass course at any point on the map.

49 The Christmas[3] drift indicator solves the velocity triangle by reference to actual ground observations suitably registered on the map.

50 Wichmann's course indicator[4] is an arrangement for drawing the velocity triangle on a celluloid coördinate field superposed on the map.

51 *Velocity-Triangle Computers.* The Campbell-Harrison and the Appleyard[5] course and distance calculators are typical instruments for solving the velocity triangle, i.e., for determining the unknown elements of Fig. 2. In the Appleyard form, time and distance can also be read off.

52 Similar results are obtained from a different mechanical movement in the Angus[6] compass course indicator.

53 German instruments of light construction are represented by the Pilehn and the Goerz[7] wind triangles, while Bennewitz[8] invented a device for this purpose to be mounted directly over the compass card.

54 Further applications of the velocity-triangle principle are found in the Leroy disk, the Bellieni computer, and the Dunoyer course indicator.[9] In the Dunoyer instrument the different speed vectors are represented by elastic threads of variable elongation, each of which is subdivided into numerous intervals to facilitate determination of times and distances.

55 While the foregoing list of more than thirty different instruments may seem bewildering, it is clear that not all the combinations of observation and computation which are possible for

[1] Eaton, pp. 19–21.
[2] Wimperis, pp. 61–64.
[3] Eaton, p. 17.
[4] Bennewitz, p. 185.
[5] Eaton, pp. 21–22.
[6] Eaton, pp. 14–15.
[7] Bennewitz, pp. 184–186.
[8] Bennewitz, pp. 187–188.
[9] Duval and Hébrard, pp. 40–42.

dead reckoning have yet been tried. For example, none of them utilizes the possibility of determining ground speed and true course solely from observations of air speed, drift, and compass bearing while flying on a straight course. This method offers favorable conditions for compass accuracy and eliminates the use of a time piece. All that is required, beyond a knowledge of the compass course, are two successive readings of the drift angle corresponding to two different air speeds; the computations being analogous to those in the case of a ship flying at constant air speed on two different compass courses.

A New Device for Ground-Speed Observations

56 Fig. 4 shows a simple device for drift observations "off the

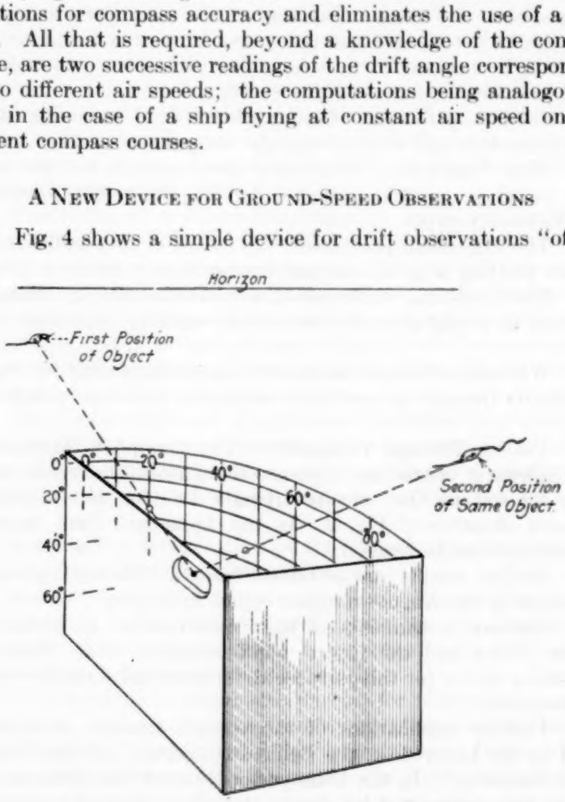

Fig. 4 Celluloid Sight for Angles off the Bow

bow."[1] It consists of a wooden box (shown with the top removed in Fig. 4) having a vertical cylindrical surface of celluloid with a radius of curvature of 5 in. This is graduated with a scale of tangents up and down, and serves to determine the bearing or azimuth of any point off the bow, and its depression below the horizontal. The box is mounted in front and to the right of the observer, so that he can sight through the hole shown in the left-hand side. It should be kept reasonably level during observations, which are

[1] Designed and constructed by R. W. Willson, 1917, together with the computing rule, Fig. 5.

taken by noticing both coördinates of any object in the field of view, at the beginning and end of a certain time interval.

57 The required time interval in seconds is equal to the height of the aircraft in hundreds of feet, measured above the level of the object sighted upon.

58 From these four observations the ground speed and drift can be determined mechanically by the computing rule shown in Fig. 5. This comprises five movable parts—A, B, C, D, and E—which in the photograph are set at the points 1, 2, 3, and 4 in accordance with the following data: Initial and final azimuth readings, 20° and 47°; first and second depressions, 25° and 35°, respectively (cf. Fig. 4). The ground speed can now be read off directly on the speed arm E at point 4, and equals 73 m.p.h. The drift is found by subtracting the initial azimuth from the drift-circle reading at point 5, so that in this example it is equal to 53−20° or 33°.

Dead Reckoning Above the Clouds

59 Sometimes, as when flying in fog or at night, none of the foregoing methods which require ground observations is applicable, and it is here proposed to develop a method of navigation applicable to such conditions, which depends upon the possibility of devising an instrument which will completely determine the component of the horizontal velocity parallel to the keel, of the actual motion of the aircraft over the ground, without observation of any object outside of the ship itself. Such an instrument was referred to in a previous paper,[1] but as there described it would require almost perfect stabilization and if fixed fore and aft would give only the keel speed, not the ground speed. The difficulties of securing the necessary stabilization are well known. Simple calculations show that a departure of θ degrees from horizontal, continuing for t seconds, will cause a cumulative error of $0.38\,\theta t$ m.p.h. in the ground-speed reading, which amounts to about 23 m.p.h. for each degree of deviation per minute. The proposed method tends to eliminate this source of error by utilizing only the difference between two successive readings of the keel-speed indicator or by permitting its readjustment in flight by comparison with an air-speed indicator.

60 The assumption will now be made that the motion of the airplane can be kept nearly uniform and straight for about a minute at a time and that the average speed and direction of the wind will be nearly constant for about the same interval. For the kind of flight here contemplated, these conditions may usually be fulfilled by careful piloting at a sufficient altitude to be above the unsteady conditions of the lower strata of the atmosphere, flying above the clouds, rather than at a lower altitude where the surface of the earth is visible.

61 *First Method.* It is obvious that if it is possible to determine

[1] Trans. A.S.M.E., vol. 42 (1920), pp. 81–118.

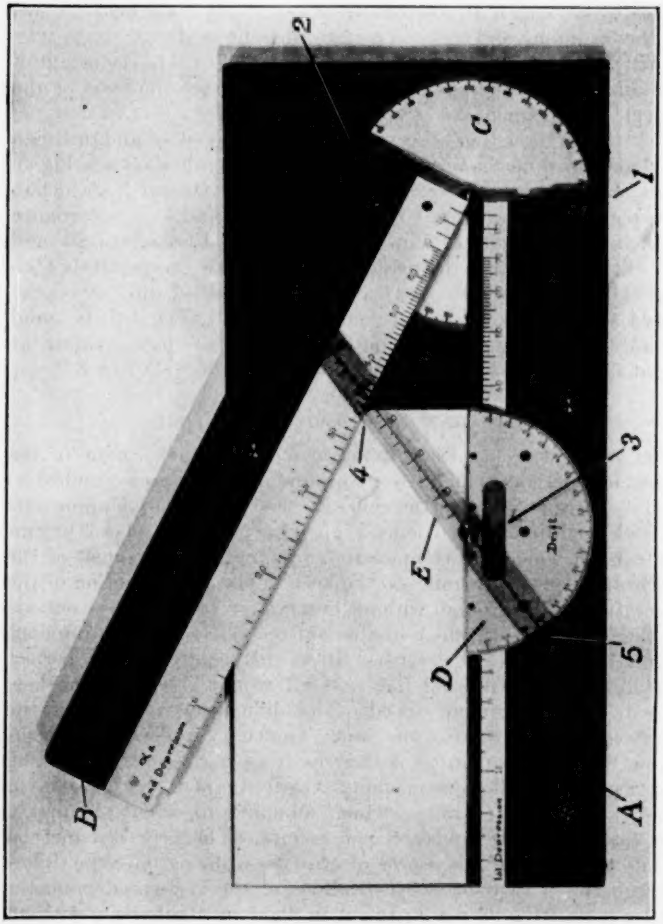

Fig. 5 Computing Rule for Angles off the Bow

TABLE 1 WIND TABLE FOR DEAD RECKONING

$$\left[\tan x = \frac{L - R}{K - A};\ W = (K - A)\sec x;\ w = a + x\right]$$

Diff. of L and R		Difference of K and A: 0	5	10	15	20	25	30	35	40	45	50	55	60
0	W =	**small**	**5**	**10**	**15**	**20**	**25**	**30**	**35**	**40**	**45**	**50**	**55**	**60**
	x (deg.) =	0/0	0	0	0	0	0	0	0	0	0	0	0	0
5	W =	**5**	**7**	**11**	**16**	**21**	**26**	**31**	**35**	**40**	**45**	**50**	**55**	**60**
	x (deg.) =	90	45	27	18	14	11	9	8	7	6	6	5	5
10	W =	**10**	**11**	**14**	**18**	**22**	**27**	**32**	**36**	**42**	**47**	**52**	**57**	**62**
	x (deg.) =	90	63	45	34	27	22	18	16	14	12	11	10	10
15	W =	**15**	**16**	**18**	**21**	**25**	**29**	**34**	**38**	**44**	**49**	**54**	**59**	**63**
	x (deg.) =	90	72	56	45	37	31	27	23	21	18	17	15	14
20	W =	**20**	**21**	**23**	**25**	**28**	**32**	**36**	**40**	**46**	**51**	**56**	**60**	**64**
	x (deg.) =	90	76	63	53	45	39	34	30	27	24	22	20	18
25	W =	**25**	**25**	**27**	**29**	**32**	**35**	**39**	**43**	**48**	**53**	**58**	**62**	**66**
	x (deg.) =	90	79	68	59	51	45	40	36	32	29	27	25	23
30	W =	**30**	**31**	**32**	**34**	**36**	**39**	**42**	**46**	**50**	**55**	**60**	**64**	**68**
	x (deg.) =	90	81	72	63	56	50	45	41	37	34	31	29	27
35	W =	**35**	**35**	**36**	**38**	**40**	**43**	**46**	**49**	**53**	**58**	**62**	**66**	**70**
	x (deg.) =	90	82	74	67	60	54	49	45	41	38	35	32	30
40	W =	**40**	**40**	**41**	**43**	**45**	**47**	**50**	**53**	**57**	**61**	**64**	**68**	**72**
	x (deg.) =	90	83	76	70	63	58	53	49	45	42	39	36	34
45	W =	**45**	**45**	**46**	**47**	**49**	**51**	**54**	**57**	**60**	**64**	**67**	**71**	**75**
	x (deg.) =	90	84	78	72	66	61	56	52	48	45	42	39	37
50	W =	**50**	**50**	**51**	**52**	**54**	**56**	**58**	**61**	**64**	**67**	**71**	**74**	**78**
	x (deg.) =	90	84	79	73	68	63	59	55	51	48	45	42	40
56	W =	**55**	**55**	**56**	**57**	**59**	**61**	**63**	**65**	**68**	**71**	**74**	**78**	**82**
	x (deg.) =	90	85	80	75	70	65	61	58	54	51	48	45	42
60	W =	**60**	**60**	**61**	**62**	**64**	**66**	**68**	**70**	**72**	**75**	**78**	**81**	**85**
	x (deg.) =	90	85	81	76	72	67	63	60	56	53	50	48	45
65	W =	**65**	**65**	**66**	**67**	**68**	**70**	**72**	**74**	**76**				
	x (deg.) =	90	86	81	77	73	69	65	62	58				
70	W =	**70**	**70**	**71**	**72**	**73**	**75**	**76**	**78**	**80**				
	x (deg.) =	90	86	82	78	74	70	67	63	60				
75	W =	**75**	**75**	**76**	**77**	**78**	**80**	**81**	**82**	**84**				
	x (deg.) =	90	86	82	79	75	72	68	64	62				

$K > A$ and $R > L$, $w = a + x$
$K > A$ and $R < L$, $w = a - x$
$K < A$ and $R < L$ $w = a + x + 180°$
$K < A$ and $R > L$, $w = a - x + 180°$

at any time the component of the ground speed parallel to the keel, hereafter called K, and the speed through the air A by the instruments used for that purpose, the speed and the direction of the wind may be found by turning through a circle of sufficient radius with uniform banking, and noting the difference of the values of K and A when the keel is pointed in different noted directions by compass.[1]

62 At two points of the circle, Fig. 6, where K and A are equal, the airplane is moving at right angles to the wind and at two points about 90 deg. removed from these points the velocity G is a maximum and minimum, respectively. From these observations, the direction of the wind (assumed to be steady) may be approximately deter-

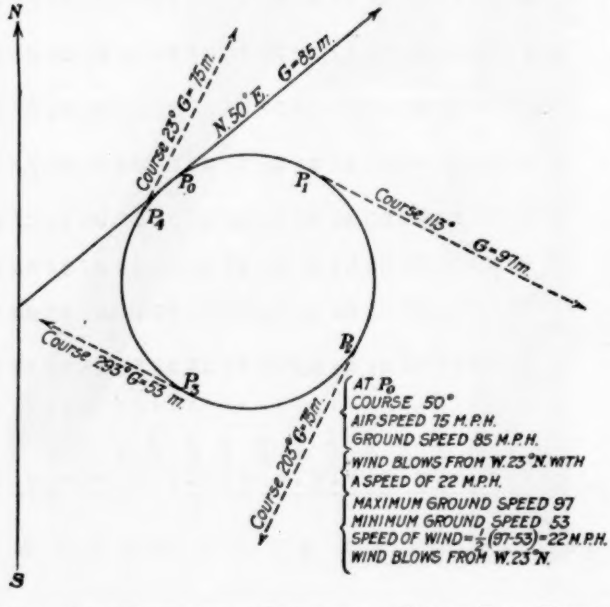

Fig. 6 Dead Reckoning by Differential Ground-Speed Readings

mined, while the difference between the maximum and minimum values of K is evidently twice the velocity of the wind.

63 *Second Method.* There are, however, some evident objections to this method which do not apply with equal force to the one now to be described, which consists in determining the direction and speed of the wind according to the following directions, in which the term "compass course" is used to signify the reading of the lubber's

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[1] From the known course and speed through the air, compounded with the velocity and direction of the wind, the true course and speed over the ground may be found by the traverse table (Fig. 9).

line of the compass; which increases from 0 deg. to 360 deg. as the nose of the airplane points from N. to E., S., W., etc.

a Note the compass course *a*, air speed *A*, and ground speed *K* parallel to the keel.

b Shift the course slowly to the right until the compass course is 30 deg. greater, and when moving steadily on this course with an air velocity *A* read the value of *K*. Call this value *R*.

c Shift the course to the left until the compass course is 30 deg. less than original course and read the value of *K*. Call this value *L*.

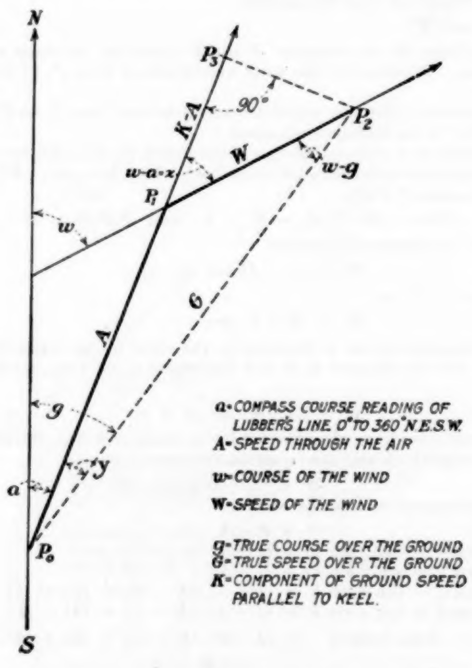

Fig. 7 Theory of Dead Reckoning by the Second Method Proposed

d Shift back to the original course *a*, and read *K* and *A*, which should be nearly the same as at the beginning.

e With the arguments $L - R$ and $K - A$ enter Table 1 and take out the values of *W* and *x*. *W* is the speed of the wind in miles per hour and $w = a + x$ is its course, i.e., the direction toward which it is moving.

f Enter the traverse table (Fig. 9) with *a* and *A* as course and distance and find the corresponding values of difference of latitudes and departure. Find in the same manner the Lat. and Dep. corresponding to *w* and *W*. Add these Lats. and Deps. and with

the two sums as Lat. and Dep. enter the traverse table and take out the true course and speed of the airplane.

The theory of the method is as follows, referring to Fig. 7 and using the notation already explained; i.e., let

K = component of the ground speed parallel to the keel
a = compass course
A = air speed
w = course of the wind
W = speed of the wind
g = true course of the airplane over the ground
G = true speed over the ground.

To find w and W:

Starting from P_0 the velocity A would carry the airplane to P_1 in the unit of time. The force of the wind would take it from P_1 to P_2 in the unit of time.

The resultant of the two velocities would take it from P_0 to P_2 in the unit of time; $P_0 P_2$ is the true ground speed.

The projection of P_0P_2 on the line of the keel is $P_0 P_3$ and is the component of the true ground speed parallel to the keel, which is equal to K.

In the triangle $P_1P_2P_3$

$$P_1P_2 = W, \quad P_1P_3 = K - A, \quad \text{and} \quad P_2P_1P_3 = w - a$$

whence the fundamental equation

$$W = (K-A) \sec (w-a)$$

or, writing $x = w-a$,

$$W = (K-A) \sec x$$

If the compass course is changed to the *right* by an angle of k deg. the value of K will be changed to R and the course to $(a + k)$, giving the equation

$$W = (R-A) \sec (x + k)$$

If the course is changed to the *left* by an angle of k deg. the ground-speed reading changes to L and the equation becomes

$$W = (L-A) \sec (x-k)$$

These three equations may be written

$$W \cos x = K-A \qquad [1]$$
$$W (\cos x \cos k - \sin x \sin k) = R-A \qquad [2]$$
$$W (\cos x \cos k + \sin x \sin k) = L-A \qquad [3]$$
$$\cos k - \tan x \sin k = (R-A)/(K-A) = [2] \div [1] \qquad [4]$$
$$\cos k + \tan x \sin k = (L-A)/(K-A) = [3] \div [1] \qquad [5]$$
$$2 \tan x \sin k = (L-R)/(K-A) = [5] \div [4] \qquad [6]$$

$$\tan x = \frac{L-R}{K-A} \frac{1}{2 \sin k}$$

and if k is 30 deg. the formula becomes

$$\tan x = \frac{L-R}{K-A} \qquad [I]$$

which determines x and therefore the course of the wind since

$$w = a + x$$

and the fundamental equation

$$W = (K-A) \sec x \qquad [II]$$

determines the speed of the wind when x has been determined by [I].

64 The practical application of the method is rendered very simple and expeditious by the use of a table of double entry giving the values of x and W with $(L - R)$ as the vertical and $(K - A)$

as the horizontal argument. Such an arrangement is given in skeleton form in Table 1.

65 It is evident that the change of course need not be restricted to 30 deg. right and left, and the same wind table may be used with any other value of k if the value $(L - R)/(2 \sin k)$ is used instead of $L - R$ as the vertical argument.

66 If, for instance, $k = 45$ deg. the argument would be very

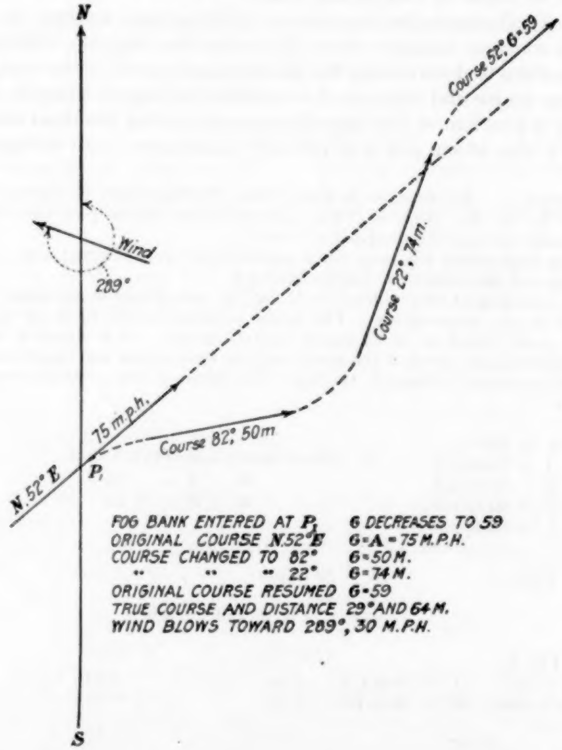

Fig. 8 Data for Illustrative Example Determining True Course and Distance

nearly seven-tenths of $L - R$, and for $k = 90$ deg. it would be exactly one-half of $L - R$. As a rule a difference of 60 deg. between the two courses is sufficient for a good determination and the loss of distance by the evolution is very small. An airplane moving at 150 m.p.h. and requiring two minutes for the two changes of course would return to her original course about two-thirds of a mile behind the place she would have reached if she had continued on the original course. If k were 45 deg., she would lose three and a half

miles and if k were 90 deg. about five miles, which is just the same loss as if she had turned a complete circle in two minutes and noted the maximum and minimum ground speeds in accordance with the first method suggested. That method, however, has the advantage of not requiring any computation or any knowledge of the air speed. A few trials in the air would show what would be the practice best suited to average conditions and what modifications should be made in exceptional cases.

67 To illustrate the convenience of the present method, we may take a concrete example which illustrates the complete solution of the problem of determining the direction and speed of the wind and the true course and distance of an airplane moving in a steady wind. This is a problem of the same kind as computing the dead reckoning of a ship at sea and is of primary importance in air navigation.

Example 1. An airplane is flying from Bolling Field to Boston on a course N. 52° E. (Fig. 8). The drift indicator shows that the wind is practically nil, and the air speed is 75 m.p.h.

A fog bank which has been noted approaching from the east now reaches the ship and the value of K falls to 59 m.p.h.

The course is at once shifted to N. 82° E. and R and A are noted as 50 and 75 m.p.h., respectively. The course is then brought to N. 22° E., and L and A are found to be 74 m.p.h. and 75 m.p.h. It is required to find the direction and speed of the wind, and the true course and speed over the ground made good through the fog. The form of the computation is as follows:

$$\begin{cases} a = 52° \\ A = 75 \text{ m.p.h.} \\ K = 59 \text{ m.p.h.} \\ R = 50 \text{ m.p.h.} \\ L = 74 \text{ m.p.h.} \end{cases}$$

ARGUMENTS FOR WIND TABLE

$$K - A = -16$$
$$L - R = -25$$

From Table 1,

$$W = 30 \text{ m.p.h.}$$
$$x = 57°$$
$$w = a + x + 180° = 289°$$

From Fig. 9,		Diff. Lat.	Dep.
$a = 52°$	$A = 75$ m.p.h.	+46	+59
$w = 289°$	$W = 30$ m.p.h.	+10	−28
Sums		+56.2	+31

Therefore the true course = 29°; ground speed = 64 m.p.h.

Example 2. Having ascertained on entering the cloud that the airplane if kept on the original course, will actually move over the ground on a course N. 20° E. with a velocity of 64 m.p.h., it would be desired to find what compass course should be steered to approach its destination as directly as possible.

This is the second important problem and may be stated thus: Given the air speed A and the course and distance of the wind, to find the compass course a which will cause the airplane to move over the ground in the desired direction. The solution is as follows:

In the triangle $P_0P_1P_2$, Fig. 3, the known quantities are A, W, and g, where g is the true course over the ground, and w is also known.

Let the angle between the true course and the course indicated by the

lubber's line of the compass be y, which evidently equals $g-a$. We may then write:

$$\frac{\sin y}{W} = \frac{\sin (w-g)}{A}$$

and

$$\sin y = \frac{W}{A} \sin (w-g)$$

which determines y (the drift angle) and therefore the course to be steered, which is

$$a = g - y$$

Also from the triangle $P_1P_2P_3$,

$$K = A + W \cos (w-a)$$

which determines K and gives a check on the computation as it tells what the keel speed indicator reading will be when on the true course.

In the problem first considered, the desired course is 52° and the data are:

$g = 52°$	$W \div A$	$= 0.40$
$A = 75$ m.p.h.	$w - g = 237°$, $\sin$	$= -0.84$
$W = 30$ m.p.h	$\sin y =$	-0.336
$w = 289°$	$y =$	$-20°$
	$g - y = a$	$= +72°$
	$w - a = 217°$, $\cos$	$= -0.80$

Therefore, $K = 75 + 30\,(-0.80) = 51$ m.p.h.

Short tables which may be printed on a single page will enable this computation to be made very quickly and with little danger of mistake.

68 There is a temptation to follow the subject further, but enough has been said to show the advantage of possessing an instrument that will give directly what may be called the keel-speed of an airplane over the ground.

Use of the Graphic Traverse Table

69 Fig. 9 shows the graphic form of traverse table[1] referred to in the preceding problems. It consists of a graduated quadrant on cardboard with a celluloid scale pivoted at the origin. Course and distance are measured along the circumferential and radial scales, respectively; difference of latitude and departure along the vertical and horizontal scales, respectively.

70 If differences of latitude and longitude are expressed in minutes, distances and departures will be given in nautical miles. This results from the fact that sixty nautical miles are identical with one degree difference of longitude on the equator or one degree difference of latitude at any point. (One nautical mile equals about 1.15 statute miles.) The advantages of the nautical mile,

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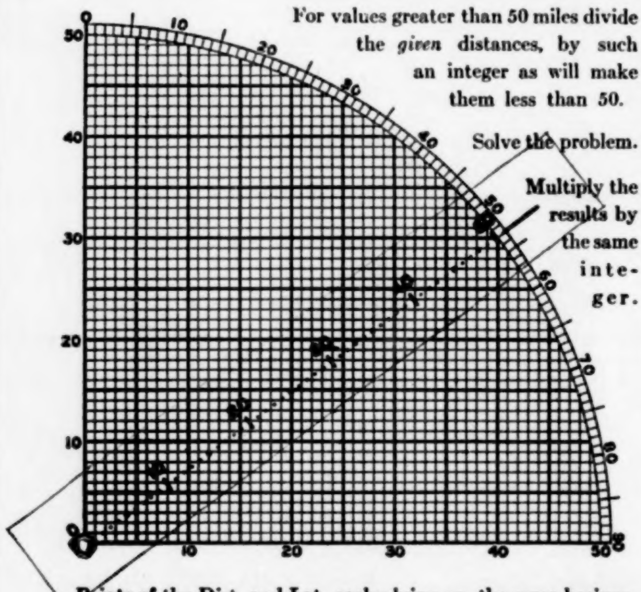
[1] From Handbook of Travel, pp. 281–385, chapter on Determining Position by Astronomical Observations, by R. W. Willson. Copyrighted. 1918. by Harvard Univ. Press.



Points of the Dist. and Lat. scales lying on the same horizontal line are "corresponding points."

Points of the Dist. and Dep. scales lying on the same vertical line are "corresponding points."

To change Course and Dist. to Lat. and Dep.; **Set the movable radius to the given course. Note the given distance on the Dist. scale. The Lat. and Dep. are the corresponding points on the Lat. and Dep. scales respectively.**

To change Lat. and Dep. to Course and Dist.: **Find a point, p, corresponding to the given Lat. and Dep. Bring the movable radius to the point, p. The reading of the radius on the divided arc is the required course and the reading of the point p on the Dist. scale is the required Dist.**

To change Dep. to Diff. Long.: **Set the radius to the Mid. Lat. Seek the given Dep. on the *Lat.* scale. The reading of the corresponding point on the Dist. scale is the required Diff. Long.**

To change Diff. Long to Dep.: **Set the radius to the Mid. Lat. Seek the given Diff. Long. on the Dist. scale. The corresponding point on the *Lat.* scale is the required Dep.**

FIG. 9 GRAPHIC TRAVERSE TABLE

which have led to its universal adoption for marine use, will be equally apparent if the nautical mile is adopted in aeronautics. In accordance with this view, air-speed indicators for long-distance flight are preferably graduated to read knots rather than miles per hour.

71 Returning to Fig. 9, it will be seen that printed instructions are provided for its use in ordinary navigation computations. In

connection with air navigation, the two following problems are of interest and may be solved as indicated below:

(*a*) *To Find the Course and Speed of the Wind:* Take $L - R$ as abscissa, $K - A$ as ordinate, and set the radial scale on their intersection. The reading of the index is x and the *course* of the wind is $a + x$.

To find the *speed* of the wind, set the index to the value of x and take $L - R$ as abscissa; the corresponding point (defined in Fig. 9) on the distance scale gives the value of W. Same directions for quadrant of x as in the margin of Table 1.

(*b*) *To Find the Course to Be Steered* when A, w, and W are given and it is desired to move on the true course g: Set the index arm to $(w - g)$ on the circle; read the abscissa corresponding to radius 50. Multiply this abscissa

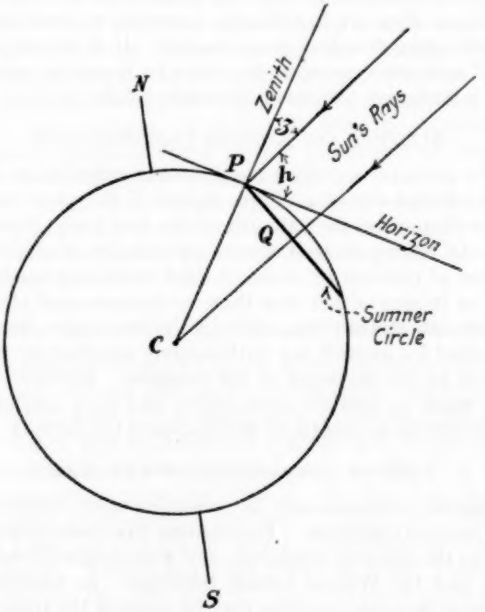


FIG. 10 SUMNER CIRCLE AND ALTITUDE OF THE SUN

by W/A (two significant figures). With this product as an abscissa, erect an ordinate. The reading on the arc where this ordinate intersects the circle is the value of the drift angle y when on the desired course, and $g - y = a$, the compass course to be set so that the true course g may be followed.

Problem (*b*) may be illustrated by the data represented in Fig. 2, where $w = 90°$ and the true course $g = 39°$, and $W/A = 30/75 = 0.40$. Setting the index arm to $90 - 39$, i.e., 51, gives 39 for the corresponding abscissa, which, when multiplied by 0.40, equals 15.6. An ordinate erected at the abscissa 15.6 cuts the arc of the quadrant at 18.3. Therefore y equals about 18° 20′, and the necessary compass bearing to make good the desired course will be $a = 39° - 18° 20'$ or about 21° east.

V—ASTRONOMICAL NAVIGATION

The Sumner Line

72 Fig. 10 serves to define the altitude h and zenith distance z of the sun (or other celestial body) as observed from any point P on the earth's surface. The locus of all points for which the altitude is the same as at P is evidently a circle of radius PQ, passing through P. The plane of this circle is perpendicular to the paper. It is known as Sumner's circle and any small arc of this circle, in the vicinity of P, short enough to be considered straight, would be called the Sumner line for point P.

73 A knowledge of the sun's altitude by sextant and of the time by chronometer enables the observer to draw the Sumner line on his chart. These data are insufficient, however, to determine at just what point along this line he is located. It is merely a "line of position," and one more such line must be found in order to get a "cut" or intersection which will definitely fix the position.

Altitude and Azimuth Determinations

74 The second observation required may either be the altitude of a second celestial object, a simple matter if the moon or stars are visible; or it may be the azimuth of the first body observed, e.g., the sun. Of course, there is always a possibility of establishing the second line of position by radio or dead reckoning (method of St. Hilaire), or in some other way than by astronomical observations.

75 Azimuth instruments, such as the theodolite, have not yet been adapted for aircraft use with entirely satisfactory results and are limited by the accuracy of the compass. Altitude determinations are made by sextant observations just as in marine practice, except for difficulties caused by the motion of the aircraft.

Types of Aircraft Sextants Available

76 Aircraft sextants may be classified with reference to the artificial horizon employed. Four groups have been developed: the gyroscopic, the damped-pendulum, and various spirit-level or bubble sextants, and the Willson bubble telescope. In addition to artificial-horizon sextants, mention may be made of the Baker sextant,[1] in which the natural horizon (cloud strata, for example) is employed, correction for dip being eliminated by combining the fore and aft horizon images, a convenient method but handicapped by the fact that cloud surfaces are not always level.

77 Gyroscopic sextants were developed at an early date by Admiral Fleuriais.[2] A recent form of gyrosextant adapted for aircraft use is the Derrien-Bonneau-LePrieur instrument.[3] Gyroscopic sex-

[1] Eaton, pp. 27–28.

[2] Proc. Naval Inst., vol. 14, p. 166, 1888; E. Caspani, Jour. de Phys., vol. 6, p. 229, 1897; Annales Hydrographique, 1904; G. M. Littlehales, Proc. U. S. Naval Inst., p. 44, Aug., 1918; A. Gray, Gyrostatics and Rotational Motion, p. 129.

[3] Duval and Hébrard, pp. 47–48; Eaton, p. 31.

tants are heavier than some types, and limited in accuracy by small errors due to vibration and precession.

78 The first instance of a pendulum sextant being used in aeronautics appears to be in connection with the Wellman-Vaniman flight[1] of 1910. A damped pendulum horizon was successfully employed during the experimental navigation flights by Prof. H. N. Russell[2] at Langley Field in 1918, but such devices are usually too heavy for aircraft use and liable to become sluggish in cold weather. It is understood, however, that a pendulum horizon, both accurate and of light weight, has been made by Pulfrich[3] of Jena; also by E. G. Fischer (U. S. Coast & Geodetic Survey).

79 A spirit-level sextant was constructed by Commander Byrd[4] for use on naval aircraft, while a sextant with spherical bubble was developed by Favé,[5] with magnetic compass attached for azimuth observations. Other bubble sextants are the German Schwartzschild[6] and the British R.A.E.,[7] in both of which a spherical bubble is employed and due recognition given to the desirability of light weight and compact form. The R.A.E. sextant is similar to the Willson sextant described below in that the images of both object and bubble move together when displaced.

80 The Willson sextant employs a bubble telescope of special design (See Fig. 11) and was described by Professor Russell at the conclusion of his experimental flights as being the most satisfactory instrument available.[8]

Details of Willson Bubble Telescope

81 The bubble telescope is a device which can be attached to any form of sextant without altering its construction. As furnished to General Kenly in 1918, it was attached to a quadrant, lighter than the usual sextant, and if desired can equally well be attached to a divided circle lighter than such a quadrant. In Fig. 12 it is shown attached to an ordinary marine sextant. In this, *B* is the bubble tube illuminated by the lamp *L* at night. The lamp *L′* illuminates the scale. These lamps are controlled by push buttons *P* attached to the handle *H*. Sun glasses *S* are placed in front of the horizon glass *G* to cut out the view through the unsilvered portion of the

[1] W. Wellman, The Aerial Age (Keller Co., N. Y.), 1911.

[2] Russell, Navigation of Aircraft by Sextant Observations, Proc. Astron. Soc. Pacific, vol. 31, pp. 129–149, 1919.

[3] Eaton, p. 30.

[4] Eaton, pp. 2, 28.

[5] Duval and Hébrard, p. 48.

[6] Eaton, pp. 2, 29.

[7] G. M. B. Dobson, *Geographical Jour.*, vol 56, pp. 370–389, 1920.

[8] Russell, loc. cit. The bubble telescope here described was developed by R. W. Willson in 1894 and shown to Professor Harkness, Superintendent of the Naval Observatory at that time, who rejected it in favor of the gyroscopic sextant without experimental trial. A later model was taken to sea for experimental tests in 1908, and this was the identical instrument employed by Professor Russell in his Langley Field flights.

horizon glass or mirror. When the sun glasses are turned down, the natural horizon is brought into the field of view of the telescope. A mirror *M* attached to the movable arm *A* reflects the image of the celestial object into the glass *G* and thence along the axis of the telescope, where it can be made to coincide with the image of the horizon, or of the bubble, by suitably adjusting the arm A. The altitude is now read on the divided circle *C*, as in the ordinary use of any sextant.

82 As shown in Fig. 12, the bubble telescope has been substituted for the original telescope of the sextant, but an alternative form has been designed which may be attached to the sextant, or preferably quadrant, without removing the other telescope. It consists of a short horizontal tube combined with a longer vertical tube and is attached beyond the horizon glass.

83 The essential principles in either case may be illustrated by Fig. 11. At night the pinhole opening 1, illuminated by a small electric lamp just above, serves as a point source of light which produces the image of a fine little star which remains at the center of

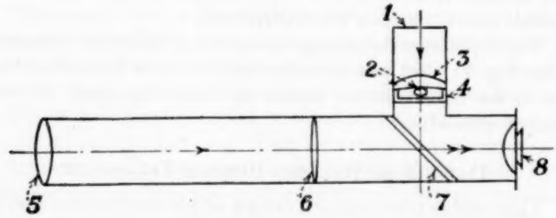


FIG. 11 WILLSON BUBBLE TELESCOPE

the bubble image. When daylight is available, the pinhole 1 is unnecessary and this part is removed together with the lamp.

84 The bubble 2 moves against the lower surface of lens 3; the liquid, usually xylol, is held in the glass box 4. Light from the sun or star under observation enters the object glass 5 and is focused in relation to the bubble image by the lens 6 before passing through the glass plate 7 into the eyepiece 8. Light from the bubble itself is reflected into the eyepiece by the upper surface of plate 7.

85 The images of the celestial object and of the bubble are superposed in the field of view, and when once brought together by the fine adjustment of arm *A* (Fig. 12), they remain together. This is true no matter into what part of the field of view the two images may be displaced by small inclinations of the sextant. Hence, it is unnecessary to go to the trouble of centering the bubble. This is an essential feature of the Willson instrument not previously possessed by any other; it is accomplished by the auxiliary lens 6, which changes the focus of the objective, so that by sliding it along the axis it makes the stellar focus exactly equal to the radius of the level, i.e., of the lower surface of the meniscus lens 3.

86 For observations by day, when arm *A* has been set to the correct altitude, one sees in the field of view a wide, dark circle surrounding a clear central disk, and the image of the sun is just enough smaller than this disk to be readily made concentric with high precision.

87 At night the problem is different, since the image of a star would not be large enough to deal with in this way. Hence, instead of illuminating the entire bubble area, a bright point of light is produced exactly at the center of the bubble image, which serves for the fiducial point much better than the bubble itself. By means

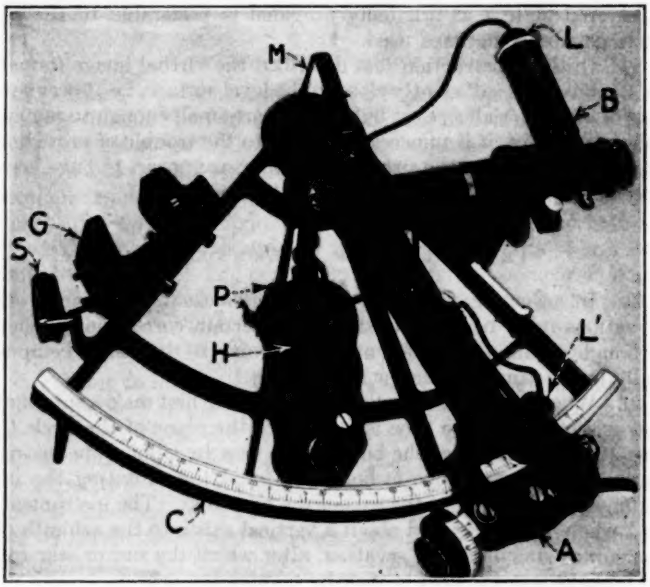


FIG. 12 BUBBLE TELESCOPE AS APPLIED TO THE SEXTANT

of arm *A* the image of the actual star is made to coincide with the artificial star, and again high precision is possible. This artificial star is produced and maintained always at the center of the bubble image by the use of some liquid of low viscosity with refractive index equal to 1.500 (xylol), and by having the bubble small enough to remain spherical. A diameter of $^1/_8$ in. is satisfactory, too small a bubble being sluggish in motion, while one too large flattens out. The bubble serves as a negative spherical lens maintaining a virtual image of the pinhole always on the level surface, i.e., on the lower surface of lens 3. This virtual image is the fiducial point.

88 The object of the meniscus lens 3 is to render the light from the pinhole 1 parallel as it passes through the bubble. Therefore

the pinhole is to be placed at the principal focus of the lens, or about 1 in. above the lens if the radius of its upper surface is $^1/_2$ in. and of its lower surface 4 in. The pinhole may be about $^1/_2$ mm. in diameter. Its function is only to serve as a point source, and not to cut down the intensity of the illumination. The intensity may be regulated in some other way, and means for doing this were provided on the quadrant as originally furnished to the Chief of Air Service. Not only is it important for the intensity of the artificial star not to be too great, but that of the actual star under observation must not be unduly diminished while passing through the optical system of the telescope; on this account the plane parallel unsilvered plate 7 as originally provided is preferable to the reflecting cube sometimes used.

89 In the construction just described the virtual image formed by the bubble is sufficiently close to the level surface, i.e., lower surface of lens 3, for all sizes of bubble that are small enough to remain spherical; hence, it is unnecessary to go to the trouble of providing means for regulating the size of the bubble, as appears to have been necessary in the case of the R.A.E. instrument.

Practical Use of Willson Sextant

90 In adjusting the bubble sextant the usual preliminary observations must be made to determine certain corrections, which, although required for different reasons, may be determined empirically in the same way as for any sextant.[1]

91 When taking observations in flight, one first makes sure that the axis of the bubble tube is parallel to the plane of the circle C, Fig. 12; he next brings the bubble into view by tipping the instrument until the telescope is horizontal, and then rotating the instrument slightly about the axis of the telescope. The instrument, as a whole, is then turned about a vertical axis into the azimuth of the sun or star under observation, after which the sun or star can be brought into the field of view by moving the arm A.

92 A special form of rolling clamp has been devised which facilitates rapid adjustment of the index arm. For aircraft observations the instrument is held in the hand in the usual manner, engine vibrations being sufficiently damped by the observer's body. To diminish acceleration errors, careful piloting is essential. These errors are not so difficult to contend with in aircraft as at sea, and may be greatly diminished by averaging observations.

93 The experimental navigation flights under the direction of Professor Russell referred to above, included more than one thousand sights on the sun, moon, and stars, and were taken from airplanes and seaplanes of nine different types. The utility of the instrument

[1] R. W. Willson, Determining Position by Astronomical Observations, loc. cit., pp. 296–301; Determination of the Altitude of Aeroplanes by Sextant Observations, Proc. Am. Acad. Arts & Sci., vol. 47, pp. 22–43, 1911.

may therefore be fairly summed up in the following quotation from his Report:

> The bubble sextant appears to leave little to be desired as an instrument for aerial navigation. It is light, portable, reasonably rugged, easy to learn to handle and to manipulate under practically all conditions, and gives a precision of observation much surpassing the limit set by the small residual irregularities in even the best piloting.
>
> Perhaps the best example of what can be done with it is found upon the last flight, when nine observations were made at night upon Sirius—the observer being in the front cockpit and having to turn around in his seat and observe behind the wings, while the ship, of type *JN4H*, was carrying two bombs, and consequently not flying easily. Nevertheless, the average error for a single setting was 14 min. and that of the mean of the nine, 2.5 min.

Reduction of Observations

94 Choice of a suitable method for rapid computation depends on the particular flight contemplated and the individual preferences of the navigating officer. Much can be accomplished by precomputation, and by choosing the most suitable map projections (Mercator, gnomonic, stereographic, Littrow, Hilleret central projection, etc.); and during the flight by graphical methods or mechanical computing devices. Examples of the latter are the line-of-position computer of Prof. C. L. Poor,[1] Bygrave slide rule, nomogram slide rule of Wimperis and Horsley, and the Baker navigating machine.[2]

Determining Compass Corrections in Flight

95 Owing to magnetic disturbances when the engine is running, final corrections of the compass, although approximately compensated on the ground, are best determined in flight. This may be done by comparing the observed compass bearing of the sun with its computed bearing obtained from astronomical tables for the locality of the ship; or in general from a knowledge of the latitude, time by chronometer, and the sun's altitude, the necessary computations having been explained in a previous publication.[3]

VI—RADIO NAVIGATION

Methods Available

96 Radio navigation by direction finder carried on the aircraft seems preferable to the older method of requesting information from shore stations. Though still undergoing development, it has been applied with considerable success.[4] A third method, used in

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[1] Poor, Simplified Navigation for Ships and Aircraft, pp. 50–71 (New York: Century Co., 1918).

[2] Wimperis, pp. 101–107; Eaton, pp. 33–41.

[3] Willson, Determining Position by Astronomical Observations, loc. cit., pp. 312–314.

[4] Eaton, pp. 41–44; F. A. Kolster and F. W. Dunmore, The Radio Direction Finder and Its Application to Navigation, Bur. Stds. Sci. Paper No. 428, pp. 529–566, 1922; L. H. Walter, Directive Wireless Telegraphy, 1921.

Germany for Zeppelin navigation, consists in the operation of directional antennae at the transmitting stations; there may be, say, 12 such antennae 30 deg. apart, or one rotating-frame aerial. The craft can find its direction by observing which signals have the greatest intensity and noting the elapsed time after the initial signal.

97 The portable types of direction finder employ a coil aerial, in which a maximum current is induced when the plane of the coil lies in the direction of propagation, and a minimum current, almost zero, when at right angles to that direction. The polar-coördinate diagram for intensity of signal as a function of angular displacement has the shape of a figure 8, so that the sharpest setting can theoretically be made when passing through the zero position. Practically, this advantage is modified by several disturbing factors, which have been surmounted in various ways described in the references quoted.

98 For such instruments an accuracy of the order of 1 deg. is claimed under favorable conditions. This does not apply when flying close along shore parallel to the coast line, or at night when all wireless communication is subject to static disturbances. It is assumed that corrections are made for influences due to the aircraft itself.

99 Whatever the form of the apparatus, it may be utilized in three ways: (*a*) For simply approaching a fixed station in a "curve of pursuit," or (*b*) for finding the drift angle by timing successive observations, or (*c*) for determining actual position on the chart by reference to two or more stations of known location. Two such stations are theoretically sufficient if their bearings are taken with reference to the compass, for in this way two "lines of position" can be drawn on the chart and the ship must be at their intersection. In view of the traditional unreliability of the compass, and on account of the desirability of an independent check, the alternative method suggested below may be of interest.

Determining Position by Radio Bearings Alone

100 The proposed method consists in determining the direction of three stations, not with reference to the compass, but with reference to the ship itself. If in this way the angle θ_1 between two stations of known location A and B is determined together with the angle θ_2 between B and C, two intersecting curves can be found on a suitably prepared chart, and this intersection represents the position of the ship.

101 Let us consider the solution in detail for the case of a plane surface, which is sufficiently accurate when the distances from the position of the ship P, to the transmitting stations A, B, and C, Fig. 13, are not much greater than 500 miles. Suppose that the radio direction finder shows the angle θ_1 formed by the great circles PA and PB to be 90 deg. and θ_2, the angle subtended between PB and PC to be 64 deg. Not knowing the compass bearing of the

ship's axis, it is impossible to lay off any of the lines *PA*, *PB*, or *PC* directly on the map; but a curve can be found passing through *A* and *B* which will be the locus of all points subtending an angle of 90 deg. between *A* and *B*. Likewise a curve can be found passing through *B* and *C* which is the locus of all points subtending an angle of 64 deg. between *B* and *C*. According to the observed data, the ship must lie on both loci at once and therefore at their intersection.

102 The points *A*, *B*, and *C* being known they can be marked on a map, the two loci can be constructed and the point *P* therefore fixed in position on the map. It only remains to discover the proper mathematical method for constructing such a curve

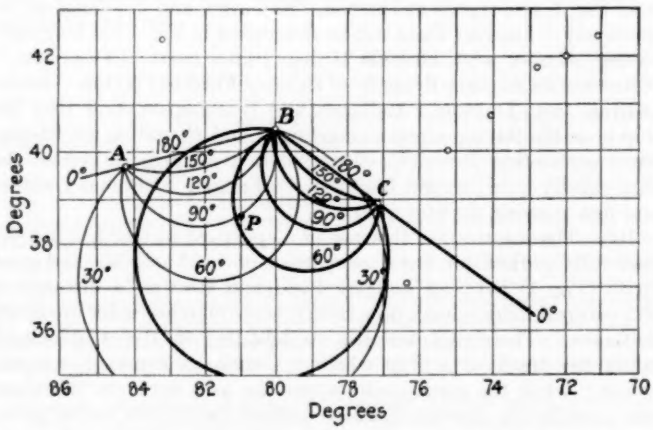

FIG. 13 DETERMINING POSITION OF AIRCRAFT BY RADIO BEARINGS ALONE

through two given points *A* and *B* and lying wholly on one side of the line *AB*, that any point *P* on the curve will subtend a constant angle θ between *A* and *B*. The solution is familiar when θ is a right angle, for in this case the locus is a semicircle, having the diameter *AB*. As the point *P* is moved around this curve from *A* to *B* the angle *APB*, designated as θ, remains constant and equal to 90 deg.

103 In general, the required locus consists of the segment of a circle whose center is so placed that the arc *AB* on the side opposite to the point *P* will be equal to 2θ, this solution being familiar in connection with the 3-point problem in surveying and marine navigation. The center of such a circle *O* is therefore located on the perpendicular bisector of the chord *AB*, at a distance from *AB* of $\frac{1}{2}\,AB \cot \theta$. The center can be located graphically by drawing a line at *A*, making an angle θ with the chord *AB* and erecting a perpendicular to this at *A*. The intersection of this perpendicular with the perpendicular bisector of *AB* is the center of the circle.

A family of circles thus can be constructed for a succession of values of θ.

104 If a chart is prepared before the flight with two sets of circles, one passing through A and B, the other through B and C, the location of the ship can be found by interpolation without further computation except such as may be needed to correct for lack of simultaneity in the radio observations. Of course, more elaborate charts can be prepared with the interval between circles reduced to 1 deg. A third family of circles constructed on the chord AC provides a mathematical check against determinations made on AB and BC, while an experimental check can be made if some fourth station D is available for observation.

105 The points A, B, and C on Fig. 13 have been transferred from Fig. 1, and represent Dayton, Pittsburgh, and Washington, respectively. Langley Field will be recognized in Fig. 13 at longitude 76 deg. 23 min. west, latitude 37 deg. 2 min. north. The point P represents an airplane flying from Langley Field to Dayton. Radio bearings from Dayton, Pittsburgh, and Washington show that the ship is on the 90-deg. circle connecting A and B, and on the 64-deg. circle connecting B and C. Interpolation on Fig. 13 locates the ship exactly on a straight line between Langley Field and Dayton, and just crossing the 81st meridian.

106 The solution for the case of a spherical surface has not yet been fully worked out, but a comparison of the rhumb line and great circle (Fig. 1) between Langley Field and Boston (a distance of 395 nautical miles) shows how trifling this difference is for moderate distances. The rhumb line is a straight line on Mercator's chart, being the track of a ship which maintains a constant compass course; while the great circle, appearing as a curve on this chart but actually the shortest distance between two points on the globe, represents the path normally followed by the radio signal.

VII—VARIOUS EXPEDIENTS FOR SAFETY AND EFFICIENCY

Meteorological Information

107 Considerable progress had been made toward the end of the war in organizing meteorological services to aid the aviator in selecting the most favorable time for departure, and in providing for atmospheric conditions which may be encountered before reaching his destination. The importance of this service for commercial aeronautics can hardly be stressed too much.

High-Altitude Flight

108 The efficiency of air navigation depends on choosing a favorable altitude, and in this respect progress seems to be continually upward; for at the higher altitudes advantage may be taken of greater wind velocities (particularly for eastward flight over the United States), the air is less "bumpy," local storms may be avoided, and

there is a proportionally greater area open to selection in case of forced landing. At altitudes greater than 10,000 ft., or for sustained flight above 5000 ft., attention must be given to the aviator's oxygen supply.[1]

Flying Over Water

109 Difficulties encountered in long-distance flight over water, aside from the risk of forced landing at some distance from help, are largely due to the uniformity of the surface of the ocean which makes it difficult to estimate ground speed. An examination of these various problems was published by J. V. Martin as early as February, 1912,[2] in the course of which a letter from one of the present authors was reproduced, from which the following extract may be of interest:

> I presume that there will be no great difficulty in making observations of the altitudes of the heavenly bodies from an aeroplane if the development of the science of aviation makes it necessary.
>
> For determining the course and distance it would be possible to learn something at any time when the airship could be made to pass nearly over some well-marked point in the water beneath—how conspicuous such objects would have to be and how frequently they would be visible in an ocean voyage you can tell better than I.
>
> I should think I had seen patches and streaks of smooth water and perhaps other objects easily visible at a mile or two of distance and sufficiently stationary to be used from a rapidly moving aeroplane, for the observations of a four-point bearing, that is, by observing the time when the object is directly beneath and again when it is left behind and depressed 45 deg. below the horizon. The distance traveled in the observed interval of time is equal to the height of the aeroplane, hence the speed may be determined while a compass bearing of the object taken at the same observations gives the course.
>
> Of course, all methods of using two bearings and the elapsed time which are useful at sea may be modified in a similar way, the problem being the reverse of the nautical problem, the distance of the aeroplane from the water being used to find the speed instead of the speed being used to find the unknown distance.
>
> It is, of course, necessary to know the height, and this requires that a reliable barometer should be carried. It has been suggested that the often unreliable aneroid be checked by finding the dip of the horizon and then computing the height; this would be possible with a fair degree of accuracy at moderate height and with a clear horizon by means of the "Navigator's Prism" described by Capt. John B. Blish, Proc. U. S. Naval Inst., vol. xxix, p. 173.
>
> At night it might be possible to drop a sodium pellet of suitable size and note the time when it has been left behind so far that its depression is 45 deg. and again when its depression is 26.5 deg. At this interval the distance traveled by the aeroplane is equal to its height above the water and the speed is thus determined. This will be evident from Fig. 1 [of the article].
>
> It would be possible, perhaps, to devise some method of marking a spot on the water that would serve as a mark in the daytime for such an observation. It is not impossible that the spray from breaking waves might under some conditions form suitable points for observation. If the ocean

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[1] F. L. Hunt, Oxygen Instruments, N.A.C.A. Rep., p. 130.

[2] Martin, Across the Atlantic by Aeroplane, *Sci. Am.*, Feb. 3, 1912, pp. 106, 16, quoting letter from R. W. Willson.

steamship routes were followed, it is probable that many ships might be used for this purpose, due allowance being made for their own proper motion.

I have considered the possibility of using a light float trailed by a fine line to indicate the direction of the aeroplane and perhaps to serve as an indicator of the speed. If the direction and force of the wind were constant at all levels from aeroplane to the water surface, or if conditions were such that all points of the line lay in the same plane, this plane would indicate pretty accurately the direction of the flight of the airship; while it is probable that conditions could be arranged so that the vertical angle at which the line left the aeroplane would give a measure of the speed. Experience would probably furnish a proper correction for the effect of cross-currents between the airship and the water surface.

It is not unlikely that only a very small float would be required and that a considerable amount of the line itself dragging through the water would furnish the necessary friction, perhaps fairly constant under various conditions of the sea. Perhaps a small patent log attached to such a line might be made to give an approximate value of the speed, or perhaps the tension of the cord as measured by spring balance would serve to measure the speed.

In any case it seems worth while to try all these methods in a set of preliminary experiments at various times and under varying conditions of wind and sea. I would suggest the following construction for the log-line apparatus: The line itself to consist of about two miles of salmon line weighing about one pound per mile and with a breaking strength of 22 lb. (these being figures which I have obtained from a commercial sample), and the height of the aeroplane to be maintained at about 3000 ft. during the observation. The accompanying diagram shows the form I should give to the log-line apparatus.

110 Several of the methods indicated in the above communication have since been put into practice, while it is believed that the salmon line still carries interesting possibilities. Illustrations may be found in the article referred to.

Night and Cloud Flying

111 Flying on a clear night presents no serious difficulty if the eyes are not made insensitive by excessive illumination of instruments.[1]

112 Powerful searchlights may become of great importance to the air navigator at night, particularly when flying over clouds. Recent developments by Sperry in this country, and others abroad, have led to promising results, and it is understood that the air mail service is establishing a complete system of searchlights and wireless beacons at intervals as close as 25 miles apart between New York and Chicago or beyond. It is hoped that detailed information on this subject may be brought out in discussion on this paper.

Landing in Fog

113 The problem of fog landing may be subdivided into the problem of landing in fog at one's destination, and that of forced landing.

114 The former presents no insuperable difficulty, but one that requires the fullest use of methods already enumerated, particularly

[1] N. E. Dorsey, N.A.C.A. Rep. 33; Bennewitz, loc. cit., pp. 221–260.

radio methods. The developments of special searchlights for fog signaling is in progress, and it has been found that certain wave lengths in the spectrum are more penetrating than others. Aerial traffic regulations are desirable, such as the restriction of each definite class of traffic to a designated altitude varying by intervals of 1000 ft. The Loth cable system[1] is of great interest as a means of guiding aircraft toward their destination. It consists of a wire connecting two aerodromes on the route, grounded at both ends, through which an alternating current of several amperes is sent. The magnetic field created in this way can be detected by comparatively light apparatus in the airplane at a height as great as 9000 ft., and, when near the ground, at a distance as far as 10 miles either side of the line.

115 The risks due to forced landing in fog over unfamiliar territory may be diminished by navigation methods which will prevent deviation to either side of the prescribed course. Therefore the worst that can happen is to be forced to descend between stations. If the stations are close together and the altitude sufficiently great at all times during the flight, it should be possible to make a safe landing with the aid of signaling devices such as have been mentioned or are under development.[2]

116 Attempts to dissipate or prevent the formation of fog have not been successful on any very large scale, although experiments of local interest have been carried out with some success on the Monongahela River at Pittsburgh by the Mellon Institute of Industrial Research. The method consists of laying down an oil film of such extreme thinness that a comparatively small quantity of oil is spread over a very considerable area. This tends to prevent fog formation over the river.

117 The upshot of all investigations toward fog prevention seems merely to point to the importance of combating the difficulties by superior skill on the part of the pilot, more reliable engine performance, and more effective use of the usual navigation and signaling devices.

VIII—INSTRUMENT PROBLEMS

Standard Instrument Equipment

118 The pilot of a typical American army plane must handle or watch numerous diverse instruments. A close-up view of the instrument board proper has been given in a previous paper together with a discussion of standard flight instruments.[3]

119 These standard instruments, excepting the power-plant group, must also be available to the navigating officer in the rear

[1] W. Loth, *Comptes Rendus Acad. Sci.*, Paris, Dec. 5, 1921; P. Franck & A. Volmerange, *l'Aéronautique*, Feb., 1922.

[2] A comprehensive study of night-flying and fog-landing methods has recently been completed by J. A. C. Warner of the Bureau of Standards.

[3] Trans. A.S.M.E., vol. 42, pp. 81–118, 1920.

cockpit (or wherever situated), and, in addition, all his various instruments for strictly navigation work. Our attention is immediately drawn to the importance of reducing the bulk of such instruments and investigating their most efficient arrangement. The navigator must be unhampered in the use of observing instruments and in the present state of the art he may wish to take on board extra instruments for experimental trial, a possibility which should be provided for.

120 A typical outfit of instruments for the navigating officer might include:

Altimeter graduated for standard air[1]
Density-corrected air-speed indicator[2] or air-log
Navy standard compass, compensated or corrected in flight
Gyro turn indicator of rather high sensitivity (a less sensitive one for the pilot)
Inclinometers and banking indicators
Ground-speed and drift indicator
Aeronautical map, on rollers, with protractor for course finding
Chronometers keeping solar and sidereal time
Bubble sextant
Computing devices according to taste
Radio direction finder
Oxygen regulator for high altitudes.

Researches and Developments Required

121 Research work on fundamental instrument problems is apt to be neglected on account of the demand for some new type of instrument from time to time. Persistent investigation of such problems should ultimately make it possible to place the design of all standard instruments on a rational instead of a cut-and-try basis, just as is the case now, for example, with the dynamo and steam turbine. For this purpose the performance of existing instruments must be more closely studied in order to fully understand the effects of temperature and vibration on the mechanism, and to develop the best methods for eliminating lost motion and friction, and for damping oscillations; materials, also, must be studied in the hope of discovering or producing alloys for springs and diaphragms which are insensitive to low temperatures and comparatively free from elastic lag.

122 A general outline of problems connected with the progressive improvement and best utilization of the standard flight instruments has been given in a recent paper[3] and it is believed that such investigations should be kept constantly going on and applied to each new class of instruments which passes from the experimental into the commercial stage.

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[1] Such altimeters are used in France to reduce the magnitude of the corrections necessary for air temperature.

[2] Such instruments are made in England. For the aerodynamic correction of venturi instruments, cf. M. D. Hersey, Prem. Cong. Int. Navig. Aer., Rap. II., pp. 79–85, 1921.

[3] M. D. Hersey, N.A.C.A. Rep. 125.

123 For navigation purposes, new inventions are in demand along the following lines: Altitude measurement without reference to barometric pressure, ground-speed measurement without dependence on visibility of the ground; absolute inclinometer uninfluenced by accelerations; geographical position indicator; and direction indicator operating independently of the earth's magnetic field. While it may be too much to expect that instruments can be devised completely fulfilling the above requirements, a number of interesting developments are reported in a recent paper[1] and, it is hoped, may be further described in the discussion to come.

IX—CONCLUSION

Technical Literature Available

124 References on air navigation and aeronautic instruments for subsequent reading may be presented in three groups: Representative general articles (by no means a complete list); detailed treatises or official reports; and bibliographies.

125 *General Articles.* As a concise introduction to the subject possibly nothing could be better than Report No. 8 of the National Advisory Committee for Aeronautics entitled General Specifications for Aeronautic Instruments.[2] At about this same period an important contribution to the earlier development of the subject appeared in a paper by E. A. Sperry on Aerial Navigation over Water[3] describing instruments invented for the purpose and showing how to interpret the appearance of the waves.

126 H. E. Wimperis[4] in 1919 gave a general account of long-distance air navigation as developed in England, while a paper of no less interest dealing more specifically with the design of instruments was published by G. M. B. Dobson[5] in 1920. A general account of the construction and testing of aeronautic instruments, based on the work of the Bureau of Standards during 1911–1919, was presented by M. D. Hersey at the St. Louis meeting of this Society[6] and, in common with the two papers above, was supplemented by a published discussion. A fourth paper of similar scope published by C. E. Mendenhall[7] in 1921 gives an account of development work by the Army Air Service during the war.

127 In forthcoming volumes of the Dictionary of Applied Phys-

[1] F. L. Hunt, N.A.C.A. Rep. 132. For precision altimeter design see also J. B. Peterson, N.A.C.A. Rep. 126, and for compass development, J. A. C. Warner, N.A.C.A. Rep. 128; P. R. Heyl and L. J. Briggs, Proc. Am. Phil. Soc., vol. 61, pp. 15–32, 1922.

[2] Second Annual Rep., pp. 25–28, 1916.

[3] Trans. Soc. Auto. Engrs., vol. I, pp. 153–165, 1917.

[4] *Aeronautical Jour.*, vol. 23, pp. 445–468, 1919.

[5] *Geographical Jour.*, vol. 56, pp. 370–389, 1920; *Nature*, vol. 106, pp. 504–506, 1920.

[6] *Mechanical Engineering*, June, 1920; Trans. A.S.M.E., vol. 42, pp. 85–118, 1920.

[7] *Jour. Franklin Inst.* vol. 191, pp. 57–86, 1921.

ics[1] we are promised articles by Comm. T. Y. Baker on Navigation and Navigational Instruments (Vol. IV), and by G. M. B. Dobson on Aircraft Instruments (Vol. V).

128 *Detailed Treatises and Reports.* Bowditch's American Practical Navigator[2] has long been the standard reference work on fundamental navigation problems. The problems of balloon navigation are fully considered in a book by A. Marcuse,[3] 1909. The treatment of Determination of Position by Astronomical Observations, by R. W. Willson,[4] referred to in previous footnotes, aims to present the essentials of astronomical navigation with all necessary tables in a minimum space. H. N. Russell's[5] Report on the Navigation of Aircraft by Sextant Observation was published in 1919 and represents the most complete investigation of that subject which has come to our attention. The compact little Primer of Air Navigation by H. E. Wimperis[6] is well illustrated by figures in the text and numerical examples to be worked, and represents the research work of the British Air Ministry. Standing in a similar relation to French experience may be mentioned the Traité Pratique de la Navigation Aérienne, by A. B. Duval and L. Hébrard.[7] Instrument equipment and navigation methods for German aircraft are fully described by K. Bennewitz.[8] The subject of radio navigation is well covered in Directional Wireless[9] by L. H. Walter.

129 Results of instrument research in the different government establishments are to be found in the Reports of the British Advisory Committee for Aeronautics, the *Bulletin du Section Technique de l' Aéronautique Militaire,* and in the Technical Reports of the National Advisory Committee for Aeronautics, Washington, D. C. Mention may particularly be made of the N.A.C.A. Reps. 125–132 inclusive, comprising a general report on the status of aeronautic instruments at the end of the war, including descriptive details of American and foreign instruments, testing methods, and experimental results. No. 131 by H. N. Eaton covers long-distance navigating instruments. This group of investigations was carried out at the Bureau of Standards mainly during 1917–1920 by a technical staff under the direction of one of the present authors, and the Reports have been found useful in compiling information for the present paper.

130 More recent papers on air navigation from many sources are

[1] Edited by Sir Richard Glazebrook; Macmillan Co., N. Y., 1922 +.
[2] U. S. Hydrographic Office, Wash., D. C.
[3] A. Marcuse, Anleitung zur Astronomischen Ortsbestimmung im Ballon; G. Reimer, Berlin, 1909.
[4] Handbook of Travel, Chap. 28, p. 385, Harv. Univ. Press, 1918.
[5] Proc. Astron. Soc. Pacific, vol. 31, pp. 129–149, 1919.
[6] Constable, London, 1920 (128 pp.).
[7] Gauthier-Villars, Paris, 1922 (60 pp.).
[8] Flugzeuginstrumente; Berlin, R. C. Schmidt, 1922.
[9] Pitman & Son, New York, 1922.

to be found in the reports of the First International Congress of Aerial Navigation held at Paris in Nov., 1921.[1]

131 *Bibliographies.* About 250 references on aeronautic instruments extending from 1892 to 1921 will be found in N.A.C.A. Rep. 125.[2] A valuable bibliography on air navigation from 1919 to 1922 (74 titles including meteorological subjects) has been issued by the Air Board, Ottawa, Canada. In his book on Directional Wireless, previously referred to, Walter gives a bibliography of about 75 titles extending through 1921.

Summary of Progress

132 The experimental navigation flights by Professor Russell at Langley Field in different types of airplanes and seaplanes, and similar flights around the British Isles in a Handley-Page have served to indicate the numerical accuracy of navigation under varied conditions. Position finding by astronomical observation may apparently be relied upon within 20 miles or better; direction finding by radio within one or two degrees under fair conditions; true course and distance by dead reckoning within about 3 deg. and 2 per cent, respectively.

133 Practical demonstration of these possibilities has been furnished by the successful flights of the *NC* boats, of Alcock and Brown, and of the *R-34* across the Atlantic; of Sir Ross Smith from London to Port Darwin, Australia, and of Coutinho and Saccadura from Lisbon to Rio de Janeiro; for there were long periods in each of these flights when the craft and its passengers were wholly dependent on dead reckoning, or radio, or astronomical navigation.

DISCUSSION

John A. C. Warner. The writer shares the authors' belief that the existing methods and devices are adequate to the needs of air navigation in practically all details. But a more complete and efficient organization of all constituent branches is essential. This involves the suitable equipment and organization of ground installations such as fields, meteorological posts, and systems of communication, and also the highly specialized training of pilots and navigators.

Among the drift devices mention might be made of the optical type recently developed in France by Barbillion and Dugit.

In connection with the sextant problem the British seem to place considerable confidence in the bubble type sextant when an artificial horizon is required, but much prefer whenever possible to use a very small marine sextant equipped with devices for rapid adjustment.

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[1] Prem. Cong. Int. Navig. Aér. Nov. 15–25, 1921; Rapports, t. I–IV, Chambre Syndicale des Industries Aéronautiques, 9 rue Anatole-de-la-Forge, Paris.

[2] Cf. Seventh Ann. Rep. N.A.C.A.

Fog-prevention experiments as conducted by the Mellon Institute of Industrial Research were conducted at an earlier date in France with much the same result. In Great Britain they have become resigned to the fog and are trying to circumvent rather than destroy it; investigation of the three most promising methods of fog dissipation, i.e., mechanical, electrical, and thermal, convinced the investigators of their impracticability.

It has been proposed[1] that one solution of the fog landing problem may lie in the selection of airdromes in pairs, the conditions around one site being customarily different from those surrounding its neighbor. If the existing conditions were favorable for valley mist an airdrome on the hill would not suffer while the site nearby might be completely enshrouded, and vice versa. Radio would offer a means of directing aircraft to the more favorable field.

HENRY N. RUSSELL.[2] Reference should be made to the great convenience of the Willson bubble sextant for aerial work. The possibility of making observations with the bubble at any point in the field of view is of fundamental importance. Successful work in the air would be practically impossible without it. In actual experience the bubble drifts about considerably, owing to the accelerations of the ship, and it is impracticable to make exact observations of the contact of two images. The best method of observation is to bring the image of the sun, moon, or star into the center of the bubble. Under the best conditions the average error of a single setting is about ten minutes of arc. This arises almost entirely from irregularities in the piloting of the air craft.

As regards the calculations the writer would like to emphasize the desirability of using the St. Hilaire method and computing in advance the expected altitude of the sun or other body to be observed, much as has been suggested by Dr. Littlehales.

With regard to instruments there is little reason for carrying both solar and sidereal chronometers. A good watch regulated to mean time and a cheap watch rated for sidereal time should suffice. The latter could be corrected at odd times during the flight with the aid of a table of comparisons prepared in advance. If the expected altitude is calculated in advance there is no need for taking a sidereal timepiece at all.

The sea horizon is better than any other when it can be seen. Level land is almost as good. A uniform layer of clouds forms a fairly good horizon, although it is not usually quite level and errors of as much as ten miles may easily occur from this cause. But even this is preferable to an artificial horizon.

ERNST G. FISCHER.[3] The devices for obtaining an artificial horizon in marine navigation appearing to be either too cumbersome or without sufficient sensitivity, the writer devised an ar-

[1] R and M No. 727, British Advisory Committee on Aeronautics.
[2] Director, University Observatory, Princeton University, Princeton, N. J.
[3] Formerly of the U. S. Coast and Geodetic Survey, Washington, D. C.

tificial horizon consisting of a pendulum of minimum mass and of maximum area opposed to a damping liquid; obtaining thereby a possible sensitivity of only a few seconds with practically "dead-beat" action.

As with this device the reflected image of the celestial object does not move with the lines defining the position of horizontality, the instrument becomes of necessity one which requires such steadiness and skill as can be acquired only by considerable experience and training.

The results of tests by experienced navigators show that in the hands of a person trained in its use, the instrument will give results with probable errors not exceeding 5 minutes on vessels of average steadiness under average conditions. The training and practice must include due consideration of the effects of "acceleration," which affects all devices of this kind excepting those controlled by gyroscopic action. With practice better results than with any of the other devices simply held in the hand and not encumbered with gyroscopic apparatus can be obtained.

G. W. Littlehales.[1] An article by the writer in the *Journal of the American Society of Naval Engineers*,[2] deals with the subject of fixing geographical position in aerial navigation by means of vessel-to-shore or vessel-to-station radio-compass bearings. This discusses Section VI of the present paper. A paragraph at the bottom of page 41 of this article contains a disclosure of the coming practical method of position-finding by long-range radio-compass bearings, instead of the imperfect and unpractical attempts that are prescribed for plotting long lines or bearing on charts, which cannot be conveniently done either quickly or accurately enough. Because the article was a discussion of the utilization of vessel-to-shore bearings, the paragraph referred to deals only with vessel-to-shore bearings by means of Hydrographic Office Publications 71 and 120 (Time-Azimuth Tables) which cover all latitudes within 70° of the Equator, but the same azimuth tables afford the best existing facilities for fixing position by means of distant shore-to-vessel radio-bearings. For example, suppose a station in latitude 23° N transmits a true bearing of 94° to a vessel in a position as yet unknown somewhere between latitudes 12° and 15° N, turning to the page for 23° of latitude in the Azimuth Tables, find 94° in the azimuth column under declination 12°, and opposite to it in the hour angle column will be IIIh. 20m. This tells us that, if the latitude of the vessel is 12°, the relative longitude, that is, the longitude counted from the known longitude of the radio station, will be IIIh. 20 m. In the same manner, find 94° in the azimuth column under declinations 13°, 14°, 15°, etc., and the corresponding relative longitudes will be found to be IIIh. 10m., IIh. 58m., IIh. 46m., etc. The foregoing proceeding is brief and simple, and

[1] Hydrographic Engineer, Navy Dept., Washington, D. C.
[2] Vol. xxxii, no. 1.

enables us to trace at once the locus of the position of the vessel; and to find its intersection with another locus traced in like manner from another station. It will be of interest to observe that the loci thus traced are arcs of great circles of the earth, whereas the loci that are similarly traced in dealing with vessel-to-station bearings are isogonic lines. By proceeding thus, we are absolved from making a particular choice of chart-projection to serve the requirements of the navigator, since the required loci may be serviceably traced with despatch on maps and charts whatever may be the laws of projection employed in their construction.

With reference to the paragraph in the authors' paper under the caption Reduction of Observations, it might prove useful to invite attention to an article by the writer entitled The Chart as a Means of Finding Geographical Position by Observations of Celestial Bodies in Aerial and Marine Navigation.[1]

To lighten the labors of the aerial navigator there is now in press[2] an extensive work by the Hydrographic Office entitled The Sumner Line of Position Furnished Ready to Lay Down upon the Chart by means of Tables of Simultaneous Hour Angle and Azimuth of Celestial Bodies, by the writer. By having available in the tables the hour angle of the celestial bodies, or the relative longitude, at successive altitudes above the horizon in latitudes one degree apart from 60° N to 60° S of the Equator, the longitude from Greenwich, on the nearest integer-degree parallel of latitude, of a point on any required Sumner line may be found by comparison of the hour angle from the tables with the corresponding hour angle of the observed celestial body at Greenwich as shown by a timepiece regulated to the meridian of that place. Through the Sumner point thus found, the Sumner line is drawn by means of the value of the azimuth tabulated in juxtaposition with the value of the hour angle.

J. G. Coffin.[3] The accuracy of the measurement of barometric pressure in rapidly moving airplanes is open to serious question. An error of nearly 1000 ft. in computed altitude is possible at a speed of 120 m.p.h., and one of 3375 ft. at 225 m.p.h.

If a body of any shape is moving rapidly through the air, there are regions on the surface of it which receive an aerodynamic pressure greater than the normal static or barometric pressure and there are also regions under small pressure than normal.

For a sphere, for example, the pressure variation is shown in Fig. 14, where radial distances outside the circle represent pressures in excess of normal and raidal distances inside represent pressures less than normal. The radius, therefore, represents true barometric pressure, but not necessarily to scale.

The difficulty cannot be met by placing the barometer in any other

[1] Proc., U. S. Naval Inst., Mar., 1918.
[2] Issued in November, 1923, as Hydrographic Office publication No. 203.
[3] Physicist, U. S. Rubber Co., New York, N. Y.

receptacle or in particular, inside the fuselage; as the outside container must also be open to the air and as before the instrument will measure the total pressure obtaining at this opening.

It occurred to the writer that this difficulty could be overcome by placing the barograph inside of a small sphere which is fully exposed to the airstream and connecting its hermetically tight case

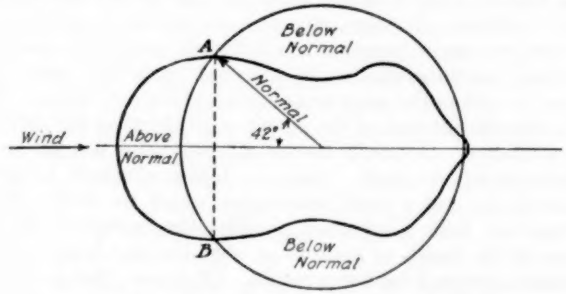

FIG. 14 PRESSURE VARIATION FOR SPHERE

to a small opening in the sphere.[1] If this sphere be now rotated by clockwork or other means at a uniform speed the opening will be carried around in a horizontal great circle and the barograph will register all the pressures, of course, passing through normal

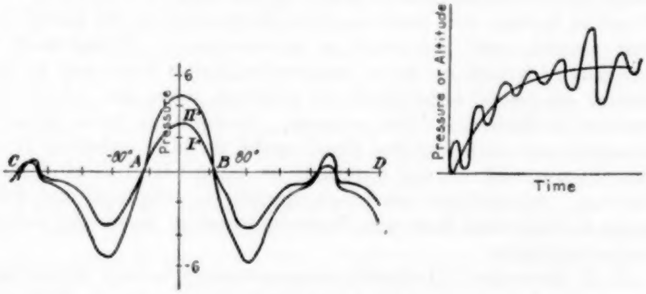

FIG. 15 BAROGRAPH RECORDS

pressure two (or more) times per revolution. The trace registered will be similar to Fig. 15.

From experiments made in the Curtiss Wind Tunnel at Garden City it is found that the zero points A and B remain fixed at all speeds experimented with. On the other hand, the change in speed produces an enlargement of the pressure scale and the pressures follow exactly the same law as if registered with a pitot tube of standard design.

[1] The barograph can, of course, also be connected to this opening by means of an airtight tube and be placed in any convenient location for observation.

It is a simple matter to calibrate such an instrument and draw the line *CD*, representing true barometric pressure, so that this troublesome problem is completely solved.

ALEXANDER MCADIE.[1] Under Instrument Problems in the standard outfit for navigation officers the authors have listed first an altimeter graduated for standard air. It is all important, of course, that an aviator should know his *exact* height above ground or sea. Ordinary altimeters give accurate elevations (so far as the correction for mean temperature of the air column is concerned) at 283 deg. absolute, that is, 10 deg. cent. or 50 deg. fahr. This, however, is seldom the mean temperature of the air column. When higher, then the altitude of the airship is greater than the indication of the altimeter. Generally the mean temperature is lower and the altimeter records too high. There are tables, of course, for getting the corrections, but a small instrument, which the writer devised, weighing less than a kilogram, enables the navigator with two motions of the hand, to read off at once his true height, that is, the height corrected for temperature. Of course, this presupposes that the navigator has some definite idea of the temperatures at the base and his own level. The instrument could probably be made to work automatically but even as it stands it would be helpful to the navigator as it does away with the use of tables for temperature correction.

C. F. MARVIN.[2] A knowledge of existing weather conditions and those to be encountered in flight is essential to aviators. The Weather Bureau now furnishes this information to the fullest extent possible with the funds at its command. Flying-weather forecasts of conditions to be expected at flying levels and at the surface are issued twice daily for fourteen zones into which the country is divided for this purpose. In addition, more detailed forecasts are made for the three model airways radiating from Washington and having terminals at Long Island, Norfolk, and Dayton. Advices also are given directly by telephone and telegraph to individual fliers who desire information personally before beginning flights.

H. E. HARTNEY.[3] It hardly seems necessary to have air navigation as accurate and precise as marine navigation, for the airplane, due to its great speed and increased maneuverability, can with little inconvenience to passengers, obliterate, as it were, an error of, say, 20 miles that might be made by relying on astronomical observations, or a two-degree error in navigation by radio finding.

C. D. HANSCOM.[4] There appear to be two distinct problems involved in the discussion of navigation: first, long-distance flights and commercial airship operation; second, ordinary commercial

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[1] Director, Blue Hill Observatory, Harvard University, Readville, Mass.

[2] Chief of the Weather Bureau, U. S. Dept. Agriculture, Washington, D. C.

[3] Washington, D. C.

[4] Chief Engineer, Glenn L. Martin Co., Cleveland, Ohio.

airplane operation. In both cases, radio direction finding would seem to be the best method available, but for commercial airplane operation, certain problems of structure and operation may perhaps be emphasized.

Owing to weight limitations, commercial airplanes must usually land for gasoline at intervals approximating 300 miles. As has been pointed out, the error even by dead reckoning will not usually be serious for this distance. A radio receiving set for direction finding would immediately correct this small error, and in thick weather a Loth cable would permit landing without difficulty. Moreover, pilots thoroughly familiar with their route, as commercial pilots should be, could pick up landmarks even in thick weather when so close to their destination.

When night flying becomes prevalent, different conditions appear. If the plane has but one engine, danger of engine failure makes it imperative that emergency landing fields should be provided at intervals of twenty miles. By placing a relatively cheap but powerful automatic beacon at each field, all navigation problems vanish; even a poor pilot can find an objective 20 miles away, especially since each beacon would be visible from the next field on ordinary nights. With twin-engined commercial planes the solution is practically the same, but for multi-motored machines it will be possible to eliminate the fear of forced landings and with it the emergency landing fields. The multi-motored plane, therefore, will normally navigate by radio direction finder, either supplemented by or supplementing dead reckoning.

A. F. Hegenberger.[1] The general impression a person will get in reading the paper by Messrs. Willson and Hersey would be that the present method of dead reckoning and astronomical navigation are very well developed, and by using the best equipment, very satisfactory results can be obtained, but our experience in the Engineering Division with a large number of the instruments referred to in the paper, does not give us that impression. We have found it is very difficult to get accurate results and determine position either by dead reckoning or astronomical navigation, under all conditions. When we arrange the conditions on which we will fly, so that they are more or less ideal, we can, by taking several observations and averaging them, determine our position within five miles. But under unusual conditions and rough weather, some of the results are quite wild, and the average would be much over 25 miles in error.

Mayo D. Hersey.[2] An interesting account of the life and scientific work of Professor Willson has been published by his associate and successor at Harvard University, Professor H. T. Stetson.[3] The death of Professor Willson occurred only a few weeks before

[1] Lieutenant, Engineering Division, Army Air Service, Dayton, Ohio.
[2] Closure to discussion.
[3] *Popular Astronomy*, vol. 31, pp. 1–5, Feb., 1923.

the annual meeting of the Society at which this paper was given. He had expected to be present, and was perfecting an experimental model of the "keel-speed indicator" for demonstration at the meeting in connection with Pars. 59–68 on dead reckoning above the clouds.

The writer is in substantial agreement with all suggestions offered in the accompanying discussion, which forms a valuable record of information and experience, especially when taken in conjunction with the discussion of aeronautic instruments at the St. Louis meeting of the Society.[1]

In addition to the fog-prevention methods cited by Mr. Warner, mention may be made of Professor W. D. Bancroft's scheme for precipitation over limited areas (e.g., landing fields) by scattering electrically charged sand from an airplane.

Lieutenant Hegenberger's reference to the practical difficulties of navigation under all but the most favorable weather conditions serves to remind us of the fact that navigation is an art as well as a science. Instrument developments alone are insufficient: there must also be extensive training and experience in the art, or knack, of using instruments to the best advantage, a matter which has doubtless received more attention abroad than it has, until recently, in this country. If a paper could be published giving the results of such experience it would be of very great help at this stage, not only for reference by practical navigators but also by laboratory investigators who are seeking to develop instruments as closely adapted as possible to actual conditions which will be encountered in the air.

[1] Trans. A.S.M.E., vol. 42 (1920), pp. 85–118

TECHNICAL COMMITTEE REPORTS

RESEARCH REPORT

PROGRESS REPORT ON THE PRESENT STATUS AND FUTURE PROBLEMS OF THE ART OF CUTTING METALS

This report places before the members of the Society a résumé of all the most important known data relating to the cutting of metals which have been developed up to 1923. It covers (*a*) design of cutting tools, including form of cutting edge and size and proportion of tool body, (*b*) materials for cutting tools, (*c*) materials used in testing cutting tools, (*d*) lubricating and cutting fluids, (*e*) historical review, and (*f*) problems for future solution.

The Special Research Committee on the Cutting and Forming of Metals which prepared this report was organized in September, 1923, by the A.S.M.E. Research Committee. Its purpose is not to carry on research of this character, but to foster such research and to act as a clearing house for all information on this subject.

Before the committee devoted its attention to the solution of any of the problems confronting it, a general discussion of the subject seemed to be desirable. It accordingly negotiated with the officers of the A.S.M.E. Machine Shop Division and it was decided that a Progress Report on the Present Status and Future Problems of the Art of Cutting Metals should be prepared by this committee and presented at the Machine Shop Session of the Annual Meeting in December, 1923. The report was reprinted in the January, 1924, issue of *Mechanical Engineering.*

PROPERTIES OF STEAM AND THE EXTENSION OF THE STEAM TABLES

The Special Research Committee in charge of this research presented its annual progress report to the Society at the Annual Meeting in December, 1923. This was published in the February, 1924, issue of *Mechanical Engineering.* It consists of (*a*) Report on the Joule-Thomson Effect, by Harvey M. Davis and R. V. Kleinschmidt of Harvard University; (*b*) Description of method by which the pressure, volume, and temperature relations for saturated and superheated steam were measured, by Frederick G. Keyes, of Massachusetts Institute of Technology; and (*c*) Design and operation of the special calorimeter, by N. S. Osborne, of the Bureau of Standards.

The progress reports covering the work of the committee during 1922 were published in the March, 1923, issue of *Mechanical Engineering.*

POWER TEST CODES

Five sections of the Power Test Codes of 1923 were completed in the year 1923. The procedure which the Main Committee on Power Test Codes has developed for the formation of these codes is long and detailed. But the committee feels that the importance of the task assigned to it justifies making sure that the new series of test codes have the approval of the Society's members in each field before they are finally issued in pamphlet

form. During the year 1923 the codes listed below were completed according to this procedure:

Test Code for Stationary Steam Boilers, Edwards R. Fish, Chairman of Committee. Tentative draft printed in January, 1921, issue of *Mechanical Engineering*.

Test Code for Evaporating Apparatus, Edward N. Trump, Chairman of Committee. Tentative draft printed in March, 1921, issue of *Mechanical Engineering*.

Test Code for Locomotives, Prof. John M. Snodgrass, Chairman of Committee. Tentative draft printed in October, 1923, issue of *Mechanical Engineering*.

Test Code for Internal-Combustion Engines, Prof. Charles E. Lucke, Chairman of Committee. Tentative draft printed in March, 1923, issue of *Mechanical Engineering*.

Code on Instruments and Apparatus: Chapter 1, General Considerations, and Chapter 2, Accuracy of Measuring Instruments, Clarence F. Hirshfeld, Chairman of Committee. Tentative draft printed in February, 1923, issue of *Mechanical Engineering*.

These five sections are being formally presented to the A.S.M.E. Council for approval and adoption as standard practices of the Society, after which they will be printed in pamphlet form.

Separate pamphlet copies of the Codes on General Instructions, Reciprocating Steam Engines, and Hydraulic Power Plants and their Equipment were made available to the membership during the year.

STANDARDIZATION

TRANSMISSION CHAINS AND SPROCKETS

The Joint Committee organized by the S.A.E. and A.S.M.E. in 1917 practically completed its work with the report on Revised Tooth Form, Space and Straddle Cutters for Sprocket Teeth and Roller Chain Nomenclature, which was published in the August, 1923, issue of *Mechanical Engineering*. In all probability this joint committee will be reorganized in 1924 as a Sectional Committee under the procedure of the American Engineering Standards Committee.

SHAFTING

The first of the three reports of the Sectional Committee on the Standardization of Shafting was published in the November, 1923, issue of *Mechanical Engineering*. This report covered standards for cold-finished shafting under the following four heads: (*a*) Standard Diameters for Cold-Finished Transmission Shafting, (*b*) Standard Diameters for Cold-Finished Machinery Shafting, (*c*) Standard Tolerances for Diameters of Cold-Finished Transmission and Machinery Shafting, and (*d*) Standard Stock Lengths for Cold-Finished Shafting.

PLAIN LIMIT GAGES

Standard Tolerances and Allowances for Machined Fits in Interchangeable Manufacture is the subject of the first report completed by the Sectional Committee on Plain Limit Gages for General Engineering Work. This set of standards was completed in the spring of 1923 and printed in abstract in the December, 1923, issue of *Mechanical Engineering*.

SCREW THREADS

In June of the present year the much-desired report of the Sectional Committee on the Standardization and Unification of Screw Threads was

presented to the S.A.E. and the A.S.M.E. as sponsor bodies. This report was printed in abstract form in the August, 1923, issue of *Mechanical Engineering*. It is based on the Progress Report of the National Screw Thread Commission. The modifications which the Sectional Committee found it necessary to make were satisfactory to the Commission and were adopted by it.

CARRIAGE-BOLT HEADS

The first standard to be submitted to the Sponsors by the Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions is the series of five "carriage" bolt heads. These are designated as (1) Regular Carriage-Bolt Head, (2) Button-Head Carriage-Bolt Head, (3) Fin-Neck Carriage-Bolt Head, (4) Ribbed Carriage-Bolt Head, and (5) Step-Bolt Head. The complete report on this standard appeared in the December, 1923, issue of *Mechanical Engineering*.

The Society is joint sponsor with the Society of Automotive Engineers for this project which is being carried forward by eight sub-committees.

SAFETY CODES

MECHANICAL POWER-TRANSMISSION APPARATUS

This year recorded also the completion of one important safety code by a sectional committee for which the Society is joint sponsor. This is the Safety Code for Mechanical Power-Transmission Apparatus, a code which provides standard rules for guarding prime movers, intermediate equipment, and driven machines. An abstract of this code appeared in the June, 1923, issue of *Mechanical Engineering*, and in July it was approved by the A.E.S.C. as a Tentative American Standard.

The Sectional Committee which drafted this safety code is sponsored by the National Bureau of Casualty and Surety Underwriters, the International Association of Industrial Accident Boards and Commissions, and The American Society of Mechanical Engineers.

No. 1916

NECROLOGY

HOWARD ELLSWORTH ADT

Howard Ellsworth Adt, president of the Geometric Tool Co., New Haven, Conn., died very suddenly on October 14, 1923. Mr. Adt was born on June 18, 1862, in Haydenville, Mass. He was educated in the public schools of Torrington. In early life he took up the study of medicine with the idea of specializing in surgery but he was persuaded to enter the business field, and in 1879 started work with John Adt & Son, New Haven.

He became very proficient in the design of special wirework and in 1880 became general designer and construction superintendent of the company. Ten years later he became general manager of the firm.

In 1895 Mr. Adt founded the Geometric Tool Co., of which he was president and treasurer. It was one of the largest concerns of its kind in Connecticut. He was also treasurer of the Bellis Heat Treating Co., a director of the First National Bank of New Haven, of the Morris Plan Bank, the Greist Manufacturing Co., and of the English & Mersick Co.

Mr. Adt was a member of the New Haven Chamber of Commerce and was very much interested in civic affairs, serving on many committees in that connection. He became a member of the Society in 1919. He belonged to the Union League and Quinnipiac Clubs of New Haven and to the masonic fraternity.

The New Haven Branch of the Connecticut Section of the Society at its regular monthly meeting passed formal resolutions of sympathy on the death of Mr. Adt, a former officer of the Section.

JOSEPH HYDE AMES

Joseph Hyde Ames, chief engineer of the American Car & Foundry Co., died suddenly in Chicago on October 11, 1923. Mr. Ames was born on March 22, 1865, in Brimfield, Mass., and received his early education at Kings School, Stamford, Conn. He started his professional career with the Northern Pacific Railway at St. Paul. In 1885 he entered the employ of the Peninsular Car Co., Detroit, as draftsman and estimator, later being made assistant superintendent.

When the Michigan-Peninsular Car Co. was formed in 1892, Mr. Ames was placed in charge of the engineering work and later became assistant general manager. In 1899, upon the formation of the American Car & Foundry Co., he was advanced to the position of mechanical superintendent of the two Detroit plants. Six years later he became chief engineer of the company in charge of improvements to plants and machinery with headquarters for some years at St. Louis and later at Chicago. During the War Mr. Ames supervised all engineering work on war contracts at the Detroit plant. After 1919 his headquarters were in New York City. He was widely known as an engineer and was considered an authority on car design and manufacturing equipment.

Mr. Ames became a member of the Society in 1898. He was also a member of the American Society for Steel Treating, the Army Ordnance Association, the Navy League, the Lawyers' Club of New York, the Chicago Engineers' Club, Old Colony Club, and the Massachusetts Society of Mayflower Descendants.

ARTHUR H. ANDERSON

Arthur H. Anderson, professor of steam and gas engineering at the University of Wisconsin, died on September 1, 1923. Professor Anderson was born in October, 1875, in Chicago, Ill. He was educated in Chicago schools and was graduated from the Armour Institute of Technology in 1902 with the degree of M.E.

The following year he worked in the gas-engine department of Fairbanks, Morse & Co., Beloit, Wis., and then became connected with the Sargent Steam Meter Co., Chicago. In 1905 he became assistant professor of mechanical engineering in the Armour Institute of Technology, where he was located for thirteen years, resigning at the end of that period to follow his profession. In 1920 he joined the engineering faculty at the University of Wisconsin, where he worked continuously until March, 1923, when he was forced to resign because of failing health. Professor Anderson became a member of the Society in 1914.

GEORGE HENRY BAYNE

George Henry Bayne, consulting engineer with the Emerson & Morgan Coal Mining Corporation, New York City, died on September 5, 1923. Mr. Bayne was born on January 29, 1881, in Baltimore, Md. He was prepared for college in St. Paul's School and Princeton Preparatory School, and then attended Cornell University, from which he was graduated in 1904 with the degree of M.E.

From 1904 to 1908 he was connected with the following firms: the Bulkly Condenser Co., Carey, Bayne & Smith Co., and Fleitmann & Co., all of New York. In 1908 he became assistant to the general sales agent of the Pennsylvania Coal & Coke Corporation, New York. Later he acted as consulting engineer for the Watkins Coal Co., and Albert A. Carey, of New York. In 1917 Mr. Bayne was commissioned a captain in the progress section of the Ordnance Department of the United States Army. Upon returning to civilian life in 1918 he became fuel expert with the Pennsylvania Coal & Coke Corporation. From 1920 to 1922 he was inspector-in-chief for the Tidewater Coal Exchange, Inc., then becoming associated as consulting engineer with the Emerson Coal Mining Corporation, New York, where he was located at the time of his death. Mr. Bayne became a member of the Society in 1911.

GEORGE BEADENKOPF

George Beadenkopf, chief engineer, gas operations, Consolidated Gas, Electric Light & Power Co. of Baltimore, Md., died on December 25, 1923. Mr. Beadenkopf was born in October, 1857, in Baltimore. He attended Baltimore City College, Maryland Institute, and Johns Hopkins University, at which latter institution he took special courses in chemistry.

In 1878 he entered the engineering department of the Consumers' Mutual Gas Light Co. in a minor capacity. He became assistant engineer of the Consolidated Gas Co. in 1883, continuing in that capacity until 1902, when he became chief engineer of the present concern, a position which he held until his death.

Mr. Beadenkopf's life and work were interwoven with every phase of the planning, construction, and operation of Baltimore's gas system, and the present plants and distribution system of that city are his work.

Mr. Beadenkopf became a member of the Society in 1912 and was one of the charter members of the Baltimore Section.

JOHN VAN VORST BOORAEM

John Van Vorst Booraem, consulting engineer and sugar manufacturer, died on May 24, 1923. Mr. Booraem was born on October 31, 1838, in Jersey City, N. J. From 1849 to 1854 he studied in France and Germany with a view to entering his father's silk-importing business. At the end of that period, however, he entered the Polytechnic Institute of Carlsruhe, Baden, from which he received his M.E. degree in 1859.

After six months' post-graduate work he returned to the United States and started work in McLeod's Steam Engine Works, Brooklyn, N. Y., as helper. He soon became a draftsman and two years later head draftsman with control over several departments. He aided in enlarging the plant and assisted in the construction of much new machinery for turning out Government contracts during the Civil War. After the war when a depression came in marine engineering, Mr. Booraem entered the employ of the Decastro & Donner Sugar Refining Co. and took charge of construction plans for a new refinery and of its erection and machinery. Upon the completion of this plant in 1872 he was made superintendent of the works, which were subsequently tripled in size under his supervision.

In 1882, when one of the Havermeyer & Elder factories was burned and the largest sugar refinery in the world built in its place, Mr. Booraem was called in as an expert to remedy an improper mounting of the centrifugals, which was completely stalling the entire plant and baffling everyone. After considerable readjustment, Mr. Booraem sucessfully overcame the trouble and was given charge of the mechanical department, while still retaining his position in the old concern. Shortly after the formation of the American Sugar Refining Co. he was made consulting engineer of the manufacturing committee and held this office for nearly ten years, meeting in weekly consultations until his resignation in 1898.

Mr. Booraem patented improved methods of washing sugar, pumping semi-solids, and washing clay deposits from light bodies, besides other inventions used by cube- and domino-sugar manufacturers. In 1906 he published a monograph entitled Internal Energy, a method proposed for the calculation of energy stored within matter.

Mr. Booraem was a life member of our Society, joining the organization in 1884. He was also a member of the American Chemical Society.

JAMES F. BOURQUIN

James F. Bourquin, vice-president in charge of manufacturing of the Continental Motors Co., Detroit, Mich., died on July 1, 1923. Mr. Bourquin was born on April 9, 1878, in Romulus, Mich. He received his early education in the schools of Detroit and then entered the University of Michigan, from which he was graduated in 1904 as a mechanical engineer.

He was employed first by the Olds Motor Works, Lansing, Mich., in experimental shop work, and then by the Chalmers Motor Car Co. in production work. Upon the reorganization of the Paige Motor Car Co. in 1910 he became its general manager, resigning five years later to aid in the formation of the Liberty Motor Car Co., of which he became vice-president.

When the United States entered the war Mr. Bourquin was called to Washington to handle the production work on government trucks, and it was during his period of service there that the class B army truck was turned out. Upon his return to Detroit he became actively associated with the Continental Motors Co., with which he was associated at the time of his death.

Mr. Bourquin was elected a junior member of the Society in 1907 and in 1914 was advanced to the grade of member. He belonged also to the Society of Automotive Engineers. He was a member of the masonic order.

WARD J. BOWER

Ward J. Bower, employed with the Goodyear Rubber Co., Akron, Ohio, died on December 7, 1923. Mr. Bower was born on September 18, 1884, in Warsaw, Ohio, where he received his early education. Later he attended the Case School of Applied Science, from which he received the degree of B.S. in 1909.

He was first employed by the Republic Iron & Steel Co., South Chicago, Ill., and then became master mechanic in the blast-furnace department of the Inland Steel Co., Indiana Harbor, Ind. During the war Mr. Bower served the U.S. Government in the Ordnance Department. After the war he accepted a position as general master mechanic with the Utah Steel Corporation and was with that concern for several years when he left to become connected with the Goodyear Rubber Co.

Mr. Bower became a junior member of the Society in 1913. He belonged to the masonic order.

CHARLES H. BRINTZINGER

Charles H. Brintzinger, appraisal engineer with the American Appraisal Co., Milwaukee, Wis., died May 17, 1923. Mr. Brintzinger was born on February 9, 1877, in Philadelphia, Pa., where he attended the grammar and high schools.

He served his apprenticeship with the Baldwin Locomotive Works from 1892 to 1896 and then became connected with the National Electric Co., Milwaukee, Wis. In 1900 he took charge of the maintenance on electric motors and cranes in the erecting and the blacksmith shop of the Allis-Chalmers Co., Milwaukee. Three years later he resigned to enter the Johnston Service Co., in the same city, to do experimental work on thermostatic heat-regulating systems. From 1905 to 1909 Mr. Brintzinger was installing engineer on air compressors and vacuum pumps for the American Air Cleaning Co. For the last thirteen years he was associated with the American Appraisal Co., as appraisal engineer, submitting valuations on manufacturing plants of every description.

Mr. Brintzinger became an associate-member of the Society in 1919.

JOHN A. BRITTON

John A. Britton, vice-president and general manager of the Pacific Gas & Electric Co., San Francisco, Cal., died on June 29, 1923. Mr. Britton had a national reputation in the gas and electric industry and was affectionately regarded by all of the fraternity as the "Dean of the Electrical Industry."

He was born on October 9, 1855, in Boston, Mass. He attended the public schools of Roxbury until he was thirteen years of age, when he moved with his family to San Francisco, continuing his education at the old Lincoln grammar school. A year later he left school to help in the support of the family, and at the age of nineteen he entered the gas business as a collector and meter reader. Shortly after that he began the study of electricity.

For many years Mr. Britton was employed by the Oakland Gas, Light & Heat Co., for which he designed the first electrical plant in Oakland. He also designed and installed their first high-pressure gas-distribution system and for fifteen years was engineer of their gas and electric departments, having supervision of all construction work and being in charge of the daily operation of plants.

In 1905, when the Pacific Gas & Electric Co. was formed, Mr. Britton became its first president, serving until 1908, when, at his own request, he

was made vice-president and general manager and as such directed the operation of this public-utility company, one of the largest in the United States.

Mr. Britton was a man of wide interests. He was active in educational work and had served as a regent of the University of California since 1903. His technical affiliations were many. He became a member of our Society in 1907. He was also a member of the American Institute of Electrical Engineers, an affiliate of the American Society of Civil Engineers, and vice-president of the National Electric Light Association. He belonged to several fraternal organizations, was a member of the Scottish Rite, a Knight Templar, Shriner and 33d degree Mason.

During the War Mr. Britton was chairman of the San Francisco chapter of the American Red Cross, and a member of the executive committee of the Liberty Loan Board and chairman of its general publicity committee. He was a director of the Panama-Pacific International Exposition. He was active in the work of the Boy Scouts and various philanthropic and charitable organizations.

WALTER BROWN

Walter Brown, general manager of the manufacturing department of the National Electric Products Corporation, Evansville, Ind., died on May 18, 1923. Mr. Brown was born on December 15, 1862, in Toledo, Ohio, where he received his early education.

His first employment was in the packing business in Toledo and later in Chicago. From 1894 to 1904 he served as secretary and general manager of the National Manufacturing Co., Elkhart, Ind., a concern which he organized for the manufacture of automatic bag machinery. For some years he was occupied in the development of the Elkhart Power Co., of which he was president and general manager. In 1910 he became sales manager for the Elkhart Manufacturing Co., where he was located for four years when he became associated with the Webster Electric Co., Racine, Wis., as vice-president and general manager in charge of the development of new lines of ignition. Early in 1922 Mr. Brown organized and incorporated the National Electric Products Corporation at Evansville, Ind., and at the time of his death held the position of general manager in charge of their manufacturing department.

Mr. Brown became a member of the Society in 1918.

CHARLES BROWNING, Jr.

Charles Browning, Jr., head of the testing laboratory, Sacramento, Cal., of the Southern Pacific Railroad, died on January 13, 1923. Mr. Browning was born in Chatham, N. Y., on March 22, 1865. He was graduated from Cornell University in 1887 with the degree of M.E.

Immediately upon graduation he entered the employ of the Southern Pacific Railroad, Sacramento, Cal., as an apprentice. From 1887 to 1892 he served as locomotive fireman and engineer, and was then promoted to the position of assistant engineer of tests. In 1905 he became chief of the testing laboratory, with direct supervision over all chemical and mechanical testing for the Southern Pacific System.

Mr. Browning's inventive genius shows in the "wheel tester" which he patented and which was designed to detect cracked and worn flanges in made-up trains. He also patented a viscosimeter which gives the absolute viscosity value for any oil. He was the author of several articles dealing with his work and published in various railroad papers. He was also the author of a book of poems entitled My Sacramento.

Mr. Browning became a member of the Society in 1905. He belonged also to a local chemists' club, the Cornell Club and to the Brotherhood of Locomotive Firemen and Enginemen.

F. L. BUNTON

F. L. Bunton, district manager of the Chicago office of the Heine Boiler Co., died on October 4, 1923. Mr. Bunton was born in August, 1868, in Elmira, Ill. He was graduated from the University of Illinois in 1891 with the degree of B.S., and then entered the employ of the Chicago & Northwestern Railroad where he served his apprenticeship.

He then became connected with the Hercules Ice Machine Co., Aurora, Ill., as erecting engineer. When this company was absorbed by the Allis-Chalmers Co., Mr. Bunton became district manager of the New York and Philadelphia offices of the Milwaukee Electric Co. Later he was connected with A. L. Ide & Sons as district manager of their Philadelphia office and then held a similar position with the Allis-Chalmers Co. He was transferred from the Philadelphia office of this company to the St. Louis office. He resigned from this firm to become district manager of the Chicago office of the Heine Boiler Co.

Mr. Bunton became a member of the Society in 1914.

EDWIN A. BURNSIDE

Captain Edwin A. Burnside, a licensed master of steam vessels and associated for forty years with the Campbells Creek Coal Co., Cincinnati, Ohio, the last twenty of which he was manager of transportation and construction engineer of the marine department, was drowned on March 16, 1922, when the steamer *Helper* capsized during a flood.

Captain Burnside was born on May 17, 1864, in Middleport, Ohio, where he was educated. He devoted practically his entire lifetime to the matter of river navigation and was credited with much success in aiding to secure the present system of locks and dams on the Ohio and Great Kanawha Rivers, thus providing all-year water transportation of coal from the great coal fields of Pennsylvania and West Virginia to the markets at Cincinnati and other down-river shipping points. He was considered the ranking authority on river navigation in the Ohio valley.

Many years of Captain Burnside's life were also devoted to the design, development and construction of improved steamboats and steam engines for western river navigation, and he was an early advocate of all-steel construction and compound-condensing engines for steamboats. He contributed many articles to the *Marine Review*, *Waterways Journal*, and other national publications relating to water transportation.

Captain Burnside became a member of the Society in 1921. He belonged also to the Society of Naval Architects and Engineers, the Cincinnati Chamber of Commerce, the American Historical Society, the National Geographic Society, and National Board of Steam Navigation. He was also one of the founders of the Naval Historical Society.

CHARLES CROOKER BUSSEY

Charles Crooker Bussey of the Bussey Process Co., New York, N. Y., died on August 29, 1923. Mr. Bussey was born on January 16, 1866, in Newburgh, Me. He received his early education in the local schools and Bucksport Seminary, with intervals of work on farm and sea. At the age of eighteen he became a second mate.

From 1885 to 1889 he served his apprenticeship with the Walworth Manufacturing Co., Boston, Mass. He soon showed marked ability in scientific research and at the age of twenty-four he started on special studies in the Massachusetts Institute of Technology. Soon afterward he entered the foundry business and was later associated with the Worthington Pump Co., and the Rand Drill Co., with whom he remained for a number of years.

In 1908 he became president of the Coal By-Products Corporation of America.

Mr. Bussey's inventions constituted a great economic advance in the method of low-temperature distillation of coals in the production of oils, coke distillates, and gas from bituminous coals, cannels, lignites and oil shales. After many years of experimenting, Mr. Bussey succeeded in bringing the process to a successful commercialization just prior to his death.

Mr. Bussey became an associate of the Society in 1918.

JAMES M. CARPENTER

James M. Carpenter, president of the J. M. Carpenter Tap & Die Co., Pawtucket, R. I., died on January 24, 1923, in Cumberland, R. I. Mr. Carpenter was born on March 28, 1838, in Cumberland, R. I., where he received his early education.

From 1853 to 1856 he served an apprenticeship as machinist and tool maker in Arnold's Machine Shop, Pawtucket, R. I. His shop experience was gained as journeyman machinist with the Corliss Engine Co., the Brown & Sharpe Manufacturing Co., and the Florence Sewing Machine Co., all of Providence, R. I. During the Civil War he worked for the Burnside Rifle Works. In 1870 he organized the J. M. Carpenter Tap & Die Co., becoming president, treasurer, and general manager of the concern, and was actively connected with this organization up to the time of his death.

Mr. Carpenter became a member of the Society in 1907. He was for many years a member of the Pawtucket Business Men's Association. He belonged also to the masonic order.

HARVEY B. CHESS, Jr.

Harvey B. Chess, Jr., president and treasurer of the Consolidated Expanded Metal Companies, Braddock, Pa., died on March 30, 1923. Mr. Chess was born on April 12, 1883, in Pittsburgh, Pa., where he attended the public schools. He prepared for college at the Shadyside Academy of Pittsburgh and then entered the University of Western Pennsylvania, now the University of Pittsburgh. He finished his collegiate work at the University of Vermont, where he received his degree in mechanical engineering.

He obtained his early shop experience in the plant of the Consolidated Expanded Metal Companies, Rankine, Pa., which was organized in 1908 and owned by his family. He served this company in various capacities until 1914, when he was made president and treasurer, which position he occupied until his death. Mr. Chess was associated with his father in the development of the earliest machinery for expanded metal lath manufacturing and personally supervised the work necessary for its progressive advancement.

Mr. Chess became a junior member of the Society in 1909 and was advanced to the grade of associate member in 1914. He belonged also to the Engineering Society of Western Pennsylvania, the Pittsburgh Chamber of Commerce, the Sons of the American Revolution and the Sons of the Signers of the Declaration of Independence, as well as to several social clubs. During the war he served in the Engineers' Corps of the U.S. Army.

CHARLES COLAHAN

Charles Colahan, inventor and mechanical engineer and a member of the Society since 1890, died at his home in Stockton, Cal., in September, 1923. Mr. Colahan was born on October 31, 1836, in Cleveland, Ohio, where he

received his early education. Later he was graduated from Child's English and Classical Institute.

Mr. Colahan had a thorough and practical experience as bookkeeper and accountant as well as buyer and salesman. For a number of years he acted as representative of New York wholesale importers throughout the middle and western states. He then organized and established the White Line Transportation Co., which was known as Colahan's Express & Fast Freight Line on the N. Y. Central road. Under Thomas A. Scott's supervision he opened the Allentown Line on the Pennsylvania road.

For fifteen years Mr. Colahan was associated with Cyrus H. McCormick as personal adviser in the construction and operation of harvesting machinery and automatic grain binders. He also had wide experience in inventing and adapting mechanical devices to various practical uses, and in this connection worked with Younglove, Massy & Co., Cleveland, Ohio, the Peekskill Plow Works, New York, N. Y., C. Aultman & Co., Canton, Ohio, and Aultman, Miller & Co., Akron, Ohio.

As advisory attorney and counsel to various large manufacturing concerns, Mr. Colahan investigated and examined into questions involving the mechanic arts and legal points relating to patentable features of mechanical devices, specializing in automatic grain binders and harvesting machinery. He was actively interested in this work up to some months before his death, when his health failed and he was forced to retire. Mr. Colahan was considered an authority in his line of work and in his death the country loses a man who did much for agriculture throughout the land.

FRANCIS J. COLE

Francis J. Cole, until recently chief consulting engineer of the American Locomotive Co., New York, N. Y., died on January 11, 1923, at his home in Pasadena, Cal. Mr. Cole was born in Rumsey, England, on May 6, 1856. His parents came to this country while he was still a child and settled on a farm in Virginia.

Farming made no appeal to the boy and he became a machine apprentice in the Mount Royal Shops of the Northern Central Railroad, Baltimore, Md. After serving his time he accepted a position as draftsman under W. H. Harrison with the Baltimore & Ohio Railroad at Newark, Ohio. He left that line to become a draftsman with the West Shore Railroad at Frankfort, N. Y., but returned after a short period to the B. & O. in Newark to accept the position of chief draftsman. A little later he was transferred to the Mount Clare, W. Va., shops of the company as chief draftsman. In 1890 Mr. Cole was appointed mechanical engineer in charge of the design of all mechanical equipment, including cars and locomotives, for the B. & O. In 1896 he resigned from this position to become mechanical engineer of the Rogers Locomotive Works, Paterson, N. J. These works were closed temporarily in 1899 and Mr. Cole accepted the position of assistant mechanical engineer of the Schenectady Locomotive Works. Later, when the American Locomotive Co. was formed, he became assistant to J. E. Sague, mechanical engineer of the company. When Mr. Sague became assistant vice-president he appointed Mr. Cole mechanical engineer, which position he held until later in life, when, wishing to be relieved of some of the arduous duties, he was appointed chief consulting engineer. This position he retained until his retirement in 1922.

During his connection with the American Locomotive Co. Mr. Cole was a leading factor in standardizing the locomotive designs and methods of the different plants which were organized under one management to form this corporation. He took a prominent part in the adoption of the superheater. His work on "boiler ratios" was a radical departure from former methods and today they are universally used.

Mr. Cole became a member of our Society in 1888. He was also a member

of the American Society for Testing Materials, the Society of Automotive Engineers, the Society for the Advancement of Science, the Franklin Institute, the American Academy of Political and Social Science, the Engineers' Club of New York, and the American Railway Association.

W. WALLACE CORE

W. Wallace Core, mechanical expert with the Borden Co., New York City, died on August 3, 1923. Mr. Core was born on January 11, 1878, in Washington, D.C. He started his technical education at the Worcester Polytechnic Institute and later completed it with International Correspondence School courses.

He served his apprenticeship with the Brown Hoisting & Conveying Co., Cleveland, Ohio, and the Baldwin Locomotive Works, Philadelphia, Pa. Afterwards he was associated for various periods with the Pennsylvania Coal Co., the New York Edison Co., the New Jersey Bridge Co. and the Monad Engineering Co. From 1905 to 1908 he had charge of the construction of pleasure machinery at Steeplechase Park, Coney Island. In 1909 he became connected with the Lidgerwood Hoisting Machine Co., in their estimating department, and a little later became superintendent of the James E. Robinson Co. From 1910 to 1913 Mr. Core was in consulting business for himself and at the end of that time became mechanical expert for the Borden Co., dealers in milk products.

Mr. Core became a junior member of the Society in 1907 and in 1914 was advanced to the grade of membership. He belonged to the masonic order.

JOSEPH V. CREEHOSKI

Joseph V. Creehoski, in charge of the engineering department of the Commonwealth Water Co., Summit, N. J., died on March 18, 1923. Mr. Creehoski was born on March 18, 1900, in Jersey City, N. J. Upon completing a mechanical course at the East Side High School in Newark, N. J., in June, 1916, he entered the drafting room of the General Electric Co., continuing his studies at the Newark Technical School, where he was enrolled in the June, 1923, class of mechanical engineering.

In 1917 he entered the employ of the Charms Co., Newark, as assistant to the supervising engineer on the installation of the dehumidifying system. Three years later he became associated with the Commonwealth Water Co., where his duties consisted of preparing plans and specifications for improvement work in connection with the installation of water mains, fire hydrants, house connections, etc.

Mr. Creehoski became a junior member of the Society in 1921.

WALTER MOLBRAY CURTIS

Walter Molbray Curtis, consulting engineer with the Leominster Shell Goods Manufacturing Co., Leominster, Mass., died on November 24, 1923. Mr. Curtis was born on October 14, 1879, in Whitman, Mass. He was educated in Massachusetts schools and was graduated from the Massachusetts Institute of Technology in 1901 with the degree of B.S.

Upon graduation he started work as detail draftsman with the Crosby Steam Gage & Draft Co., Charlestown, Mass., and a year later became head designing draftsman. From 1905 to 1907 he served as instructor in mechanical engineering in Union College, Schenectady, N. Y., then becoming assistant professor of mechanical engineering at the University of Maine. Four years later Mr. Curtis became a member of the firm of the New England Audit Co., and manager of the engineering department. He resigned from this position in 1914 to become consulting engineer with the

Fiberloid Corporation, accepting, in 1922, in addition to his duties in this position, the treasurership of a subsidiary concern, the Yale Novelty Co. Upon the sale of this company, Mr. Curtis became consulting engineer with the Leominster Shell Goods Co., where he was located at the time of his death.

Mr. Curtis became a member of the Society in 1912. He belonged to the masonic fraternity, was a Shriner, and was also a member of several social clubs and organizations.

CLARENCE ELBERT DEPUY

Clarence E. DePuy, professor of mechanical engineering at Lewis Institute, died on August 28, 1923. Professor DePuy was born on March 11, 1863, in Jackson, Mich., where he received his early education. After being graduated from high school he worked for five years in the Dennis Machine Shop of Jackson and then entered Cornell University. He finished his course at the University of Michigan, receiving the degree of B.S. in 1891.

In that same year he became a teacher in the Chicago Manual Training School. Five years later he became associated with the Lewis Institute as head of the mechanical department.

Professor DePuy took a prominent part in the civic activities of Oak Park, where he resided. He became a junior member of our Society in 1891 and in 1903 was advanced to full membership. He was also a member of the American Society of Steel Treaters and the Society of Automotive Engineers.

HARRY EARL DEWEY

Harry E. Dewey, sales engineer with the Canadian Branch of Allis-Chalmers, Ltd., died on May 22, 1922, in Montreal, Que., Canada. Mr. Dewey was born on May 7, 1878, in Ravenna, Ohio, where he received his early education. Later he attended Case School of Applied Science from which he was graduated in 1906 with the degree of B.S.

Before entering college Mr. Dewey served an apprenticeship as a mechanic. Upon being graduated he took charge of the outside erection of refrigerating units. For three years he served as a traveling expert overseeing the original installation of locomotive devices. From 1912 to 1915 Mr. Dewey worked on the setting of the lock-operating machinery on the Panama Canal. From 1915 to 1917 he had full charge of the installation of producer-gas and foundry equipment. He was for a short period associated as salesman with the Smith Gas Engineering Co., Dayton, Ohio, and at the time of his death held a similar position with the Canadian Branch of Allis Chalmers, Ltd.

Mr. Dewey became an associate member of the Society in 1920. He belonged to the masonic order.

HORACE G. DIMICK

Horace Greeley Dimick, vice-president and chief engineer of the O'Keefe-Orbison Engineering & Construction Co., Appleton, Wis., died suddenly on December 18, 1923. Mr. Dimick was born on May 18, 1868, in Berlinville, Ohio. He was graduated from the Western Reserve Normal School of Milan, Ohio, in 1888. Later he supplemented this training by study in electrical, mechanical and civil engineering.

In 1889 he went to Chicago to enter the employ of the Western Electric Co. He remained with that concern until 1905, working through many different departments, and then left to become associated with the O'Keefe-Orbison Engineering & Construction Co. At the time of his death he was vice-president, general manager and chief engineer of the firm.

Mr. Dimick became a member of the Society in 1921.

ALEXANDER GETCHELL DRURY

Alexander Getchell Drury, mechanical and electrical engineer of the Cincinnati, Newport & Covington Street Railway Co., Cincinnati, Ohio, died on November 28, 1923. Mr. Drury was born on June 27, 1885, in Cincinnati, where he received his early education. He was graduated from Cornell University in 1902 with the degree of M.E.

At that time he entered the employ of the Bullock Electric Co. where he gained his shop experience. A year later he became connected with the gas-meter department of the City of Cincinnati. Since 1913 to the time of his death, Mr. Drury held the position of mechanical and electrical engineer of the Cincinnati, Newport & Covington Street Railway Co.

Mr. Drury became a member of the Society in 1921. He belonged to the masonic order in which he was actively interested and to the Ohio Sons of the Revolution. He was a frequent contributor to the technical press and was co-author of a book on X-Ray Operations.

THOMAS P. EGAN

Thomas P. Egan, president and founder of the J. A. Fay & Egan Co., Cincinnati, Ohio, died on January 10, 1922. Mr. Egan was born on November 20, 1847, in Limerick, Ireland. While but a child he was taken by his parents to Hamilton, Ont., Canada, where he was educated. At the age of sixteen he secured work in a machine shop in Cincinnati and after a few months there, changed to the firm of Steptoe, McFarlan & Co., which at that time was one of the pioneer manufacturers of woodworking machinery in the United States.

He remained with that concern for twelve years. Early in his employment he had the misfortune to lose his right arm and because of this accident he was put to work in the office of the firm where he studied bookkeeping and the technique of the business. In 1874 Mr. Egan, with two partners, opened a little shop and seven years later the Egan Co. was incorporated with a capital of $150,000 with Mr. Egan as its president. In 1893 this company was consolidated with J. A. Fay & Co., then the most important woodworking establishment in the country.

Mr. Egan became a member of the Society in 1903. He was actively interested in the citizens' organization of Cincinnati and in 1908 was elected president of the Chamber of Commerce. He was the organizer and first president of the National Association of Manufacturers and in 1900 was awarded the decoration of the Legion of Honor by the French Government because the woodworking machinery which he had designed had proved of such value in the development of that industry in France.

ALEXANDRE GUSTAVE EIFFEL

Alexandre Gustave Eiffel, engineer and scientist, builder of the famous Tower which bears his name and author of scientific treatises of inestimable value, died on December 28, 1923.

M. Eiffel was born at Dijon, France, on December 15, 1832. After his graduation from the Ecole Centrale des Arts et Manufactures in 1885, he became interested in the study of construction in steel. At the age of twenty-six he was intrusted with the construction of a steel bridge at Bordeaux. In the foundation of this bridge he was one of the first to make use of compressed air. A little later he applied an improved system of sinking hollow piles with compressed air and the aid of a hydraulic press.

In 1867 Eiffel founded an establishment for iron and steel construction which was operated under his direction until 1890. In assembling his structures he introduced many noteworthy improved methods, through which he succeeded in placing with boldness and precision trusses and

bridges of lengths which were then quite exceptional. He improved the placing of straight bridges by devising a tipping frame and applying it to spans where this method was thought impracticable. He invented and made popular demountable and portable bridges for military and colonial use.

Among Eiffel's great works may be mentioned the viaduct of the Oise, the railway bridge over the Tarde, the railway and carriage bridges over the Tagus and at Vianna in Portugal, a number of bridges in Cochin-China, the great arched bridge at Szegedin in Hungary, the bridge at Oporto over the Douro, and the bridge of Garabit, one of the most remarkable works of the kind in France and probably Eiffel's most important work. Among other constructions he has to his credit are numerous steel-frame buildings, including churches, railway stations, and public buildings. For the dome of the equatorial telescope of the Observatory at Nice, Eiffel was awarded the Monthyon Prize in Mechanics by the Académie des Sciences.

In 1887 the Panama Canal Company intrusted to Eiffel the general plan for a system of locks. He performed this task in a way which should have assured the accomplishment of the great enterprise, but the work was stopped by the unfortunate bankruptcy of the company. In the investigation which followed M. Eiffel was fully exonerated from complicity in any of the corrupt transactions which wrecked the enterprise. The injustice of the attacks on him was demonstrated by the Court of Cassation from the judical point of view and by the Council of the Legion of Honor from the moral point of view.

Eiffel's best-known work is the Eiffel Tower which was constructed as one of the features of the Paris World's Fair in 1889. The Tower is 984 feet above the ground, contains 7000 tons of iron in its construction, commands a view of 85 miles, and houses a fine meteorological observatory with physical and biological laboratories. The Tower was constructed at a cost of more than a million dollars for which the French government voted only $292,000. Eiffel himself supplied the remaining three-quarters of a million dollars, for which he was granted the receipts from admission fees for a period of twenty years. The receipts for 1889 alone all but reimbursed him.

Since 1890 M. Eiffel devoted himself entirely to scientific research, using the Tower observatory and laboratories for the purpose. He published numerous scientific works on meteorology. He made studies of the problems of wind pressure and currents and contributed in large measure to the progress in aviation through his experiments and researches in aerodynamics. His great work, Resistance of the Air, published in 1913, was immediately translated into English and is one of the most important contributions ever made to this problem.

M. Eiffel was elected an Honorary Member of the American Society of Mechanical Engineers in 1889. He is a Past-President of the Société des Ingénieurs Civils de France and an honorary member of most of the great scientific societies of Europe and America. He was a laureate of the French Institute, an officer of the Legion of Honor, a Knight of the Iron Crown of Austria, and a Commander of the Order of the Conception of Portugal, of the Crown of Italy, of Isabella of Spain, and of St. Anne of Russia.

LEWIS M. ELLIS

Lewis M. Ellis, who was engaged in research work on the high-pressure automotive steam boiler at the Mason Laboratory of Yale University, died on December 13, 1923. Mr. Ellis was born on June 18, 1867, in Connersville, Ind., where he attended the public schools. Later he took a course at a business college in Cincinnati.

He was first associated with the Nordyke & Marmon Co., Indianapolis,

Ind., in charge of production and costs systems. From 1905 to 1908 he was with the Gray Motor Co., Detroit, Mich., as general manager. He resigned from this position to become general manager of the Ellis Engine Co., also of Detroit. From 1913 to 1918 Mr. Ellis engaged in efficiency work for several concerns, then installing for the Buick Motor Co. their personnel service department. In 1920 he became director of industrial relations with the Overland Motor Co., Toledo, Ohio, and the following year general manager of the Winslow Safety High Pressure Boiler Co., Chicago.

For the two years preceding his death Mr. Ellis was engaged in designing and testing a high-pressure steam boiler of a new type, compact in size and adaptable to automotive-truck usage. Associated with him was C. C. Trump of the Stumpf Una-Flow Engine Co., Syracuse, N. Y. All designing work, tests, etc., were conducted in Mason Laboratory of Yale University with students assigned as helpers.

Mr. Ellis became an associate of the Society in 1923. He belonged to the masonic order, the Knights of Pythias and the Odd Fellows.

ROBERT JOSEPH FLEER

Robert J. Fleer, heating and ventilating engineer and sales agent for the R. A. Dunham Co., Long Island, N. Y., died from accidental drowning on December 1, 1923. Mr. Fleer was born on May 4, 1896, in Brooklyn, N. Y., where he attended the Boys' High School. Later he entered Stevens Institute of Technology and was graduated in 1918 with the degree of M.E.

During the War Mr. Fleer served as lieutenant in the U.S. Army Transport Division. Upon being discharged he became connected with the Texas Co. as lubricating engineer. Later he was with the Sinclair Oil Co. as fuel engineer, resigning to become sales engineer with the Johns-Manville Co. At the time of his death he was associated with the R. A. Dunham Co.

Mr. Fleer became a junior member of the Society in 1919.

WILLIAM FORSYTH

William Forsyth, who became a member of the Society in 1883 and who served as manager from 1888 to 1891, died on December 18, 1923. Mr. Forsyth was born on July 2, 1852, in Northumberland, Pa., where he received his early education. Later he studied for one year at the Polytechnic College of Pennsylvania at Philadelphia.

He entered the railway service in 1869 as a machine apprentice in the employ of the Philadelphia & Reading Railway, then becoming a draftsman with the same road. Later he was connected with the Altoona Iron Co., the Pennsylvania Railroad and the Pittsburgh, Fort Wayne & Chicago Railway, becoming in 1882 mechanical engineer with the Chicago, Burlington & Quincy Railroad at Aurora, Ill. After sixteen years in this position he resigned to become superintendent of motive power of the Northern Pacific Railway at St. Paul, Minn. Mr. Forsyth served for the year of 1901–1902 on the faculty of Purdue University as associate professor of locomotive and car design. From 1903 to 1913, when he retired from active affairs, he held the position of mechanical engineer with *Railway Age*, Chicago.

CHARLES L. FREDERICK

Charles L. Frederick, vice-president and general manager of the Passaic Metal Ware Co., Passaic, N. J., died on November 16, 1923. Mr. Frederick was born in March, 1883, in Owensboro, Ky., where he was educated. He served his apprenticeship with the Owensboro Wheel Co., and then became a machinist in the employ of the Packard Motor Car Co., Detroit.

From 1907 to 1913 Mr. Frederick was connected with the New York

City office of the Packard Motor Car Co., first as shop superintendent, and then as manager of various departments requiring supervision of erection of buildings, equipment and operation of machine shops. From 1913 to the time of his death, he was associated with the Passaic Metal Ware Co., as vice-president and general manager, with supervision of all manufacturing. Mr. Frederick became a member of the Society in 1917.

E. CLAYTON GOODWIN

E. Clayton Goodwin, treasurer of the Hart & Cooley Co., New Britain, Conn., died on September 5, 1923. Mr. Goodwin was born in 1886 in Terryville, Conn., where he was educated. In 1883 he started work with the Eagle Lock Co. of Terryville, serving a three-year apprenticeship as machinist. He was connected with this firm until 1901, designing and superintending the construction of automatic die and tool work and supervising the standardizing of jigs, tools and fixtures for combination die work.

In 1901 Mr. Goodwin became associated with the Hartley & Cooley Co., New Britain, as mechanical engineer in charge of the production of wrought-steel registers, and with the entire mechanical end of the business under his personal supervision. In 1903 he became superintendent of the plant and in 1907 was elected a director. Two years later he became treasurer of the firm, which position he held at the time of his death. He was also a director of the Hart & Hutchinson Co., and a director and vice-president of the Fafnir Bearing Co.

Mr. Goodwin became a member of the Society in 1915. He was formerly chairman of the New Britain branch of the Connecticut Section of the Society. He belonged to a number of clubs and was greatly interested in civic affairs, serving on a local school board, the republican town committee and for one term as representative in the general assembly.

WILLIAM GEORGE GOODWIN

William G. Goodwin, chief engineer and superintendent of the water department of Kansas City, Mo., died on November 21, 1922. Mr. Goodwin was born on October 10, 1861 in Oceola, Ark. He was educated in the common and high schools on Winsfield, Tex.

After serving an apprenticeship in the machine shop and round house of the Texas & Pacific Railway Co., Mr. Goodwin entered the employ of the Houston Water Works Co., Houston, Tex., where for three years he served as assistant engineer. At the end of that period he became chief mechanical engineer of the Cable Railway Co., Kansas City, Mo., resigning to take the position of chief engineer with Hall Brothers in that city. From 1900 to 1910 he served Kansas City as chief engineer and superintendent of the water department and then entered into the consulting field where until 1918 he was principally engaged in designing and constructing water works, light and power plants. In 1918 he again became chief engineer and superintendent of the water department, which position he held at the time of his death.

Mr. Goodwin became a member of the Society in 1919.

WILLIAM EWART GOUGH

William Ewart Gough, mechanical engineer with the Bassett Metal Goods Co., Derby, Conn., died on February 28, 1923. Mr. Gough was born on September 5, 1882, in South Wales, England. He was brought to this country when he was still a child and was educated in the schools of Shelton, Conn.

At the age of eighteen he entered the employ of the American Brass Co.,

Ansonia, Conn., to serve his apprenticeship in the drafting room, where he remained six years. At the end of that period he became draftsman for the Farrell Foundry & Machine Co., Ansonia. Two years later, in 1906, he became connected with the Birmingham Iron Foundry, Derby, Conn., as a designer of rubber machinery. From 1908 to 1914 Mr. Gough was assistant chief draftsman with the Whitlock Printing Press Co., resigning then to become division overseer on cartridge and bullet production tools with the Winchester Repeating Arms Co. in New Haven, Conn. Four years later he became associated with the Bassett Metal Goods Co. as chief draftsman, shortly being advanced to the position of assistant mechanical engineer. In 1919 he was appointed mechanical engineer for the company, which position he held at the time of his death.

Mr. Gough became an associate member of the Society in 1920. He belonged to the masonic order and to the Elks as well as to several other fraternal organizations.

LEVI RAWSON GREENE

Levi R. Greene, consulting engineer since 1869 for the Walworth Manufacturing Co., Boston Mass., died on June 4, 1923. Mr. Greene was born on October 8, 1832, in Sutton, Mass. In 1850 he began an apprenticeship with the Corliss Steam Engine Co., Providence, R. I. Seven years later he entered the United States Navy as third assistant engineer. He was promoted to second and later to first assistant and served on the U.S. vessels *Saranac, Niagara, Adelaide and Georgianna, Massasoit, Brooklyn,* and *Shamokin.* In 1869 he resigned from the Navy to become consulting engineer for the Walworth Manufacturing Co. He remained with this concern until his death.

During the many years that the firm was concerned with the plans and fittings of large public works, Mr. Greene, as consulting engineer, was immediately interested in the construction of buildings all over the eastern part of the country, from the Parliament buildings in Ottawa, Canada, to the State buildings in Texas.

Mr. Greene became a member of the Society in 1893. He belonged to the masonic order.

G. A. GULOWSEN

G. A. Gulowsen, draftsman with the Babcock & Wilcox Co., Bayonne, N. J., and a member of the Society since 1897, died on July 28, 1923. Mr. Gulowsen was born in 1860 in Norway, where he was educated. He was graduated from a technical school there in 1878 and then served a three-year apprenticeship in that country.

In 1882 he came to the United States where he became a draftsman with the S. S. White Dental Manufacturing Co., Staten Island, N. Y. Five years later he became connected with the Torsion Balance Co., Jersey City, N. J. In 1888 he became a draftsman with the Midvale Steel Co., Philadelphia, Pa., and in 1890 accepted a similar position with the Manufacturing Investment Co., Madison, Me. From 1893 to 1895 he served as designer with the S. S. White Dental Manufacturing Co., and then became a designer with the Simonds Rolling Machine Co., Fitchburg, Mass. In 1910 Mr. Gulowsen became associated with the Babcock & Wilcox Co., as draftsman.

ROBERT H. HACKNEY

Robert H. Hackney, organizer and for twenty years president of the Pressed Steel Tank Co., Milwaukee, Wis., died on February 4, 1923. Mr. Hackney was born on August 6, 1870, in Milwaukee, Wis., where he received

his early education. Later he attended the University of Wisconsin from which he was graduated in 1893 with the degree of B.S.

For the first three years upon graduation he worked as a draftsman in the Pressed Steel Car Co., Chicago, Ill., and then for two years served as shop foreman. In 1899 he became superintendent of the Joliet works of the Pressed Steel Car Co. of Pittsburgh, Pa., where he remained three years. In 1901 Mr. Hackney went to Milwaukee where he purchased the assets of the Seamless Structure Co. and organized the Pressed Steel Tank Co., a Wisconsin corporation. He continued at the head of this company directing its growth until it became one of the most productive industries of that city.

Mr. Hackney became a member of the Society in 1922. He was a member of the board of directors of the Merchants' and Manufacturers' Association of Milwaukee, the Metal Works Association of Milwaukee, and a number of other organizations and clubs in that city.

IRVING G. HALL

Irving G. Hall, designer with the Landis Engineering & Manufacturing Co., Waynesboro, Pa., died in Bridgeport, Conn., on March 22, 1923. Mr. Hall was born on May 7, 1897, in Waterbury, Conn. He was educated in the public schools of Danbury and Bridgeport.

His first employment was with the Crane Valve Co., Bridgeport, where he served an apprenticeship as a draftsman. Later he was connected for short periods with the Remington Arms Co., the American Graphophone Co. and the Bullard Machine Co., all of Bridgeport, and in all of which his work had to do with designing and drafting. In 1917 he became draftsman and designer for the Yale & Towne Manufacturing Co., Stamford, Conn. Two years later he became associated with the Landis Engineering and Manufacturing Co. where he designed tools for electric program machine clocks.

Mr. Hall became a junior member of the Society in 1919. He belonged to the masonic order.

GEORGE DANIEL HECK

George D. Heck, designer employed by the S. S. White Dental Manufacturing Co., Prince Bay, Staten Island, N. Y., died on April 8, 1923. Mr. Heck was born on March 31, 1865, in Albany, N. Y., where he received his early education. Later he attended the Kursteiner and White Institute, a preparatory school.

From 1881 to 1886 he served an apprenticeship as a machinist in the printing-press works of R. Hoe & Co., New York City. During that period he attended a private night school to study mathematics and mechanical drawing.

Mr. Heck entered the employ of the S. S. White Dental Manufacturing Co. as draftsman in 1887. He was soon intrusted with important work and at the time of his death was in responsible charge of the design and construction of dental equipment on which he was an acknowledged consulting authority.

Mr. Heck became a member of the Society in 1916.

RUDOLPH HERING

Rudolph Hering, consulting hydraulic and sanitary engineer of New York City and an authority on municipal water supply, sewage, and garbage disposal, died on May 30, 1923. Mr. Hering was born in Philadelphia, February 26, 1847. He was educated in Dresden, Germany, and was

graduated from the Royal Polytechnic College of Dresden as a civil engineer in 1867. He returned to this country shortly afterward and became assistant engineer to the Board of Public Works of Philadelphia. In 1881 he visited Europe again to study sewage-disposal methods for the National Board of Health.

From 1883 to 1885 he was engineer in charge of the new water supply of Philadelphia, and during the next two years was chief engineer of the Chicago Drainage and Water Supply Commission. From 1888 until 1920, when he retired from active practice, Dr. Hering was engaged in consulting work for water supply and sewage works for practically all of the larger cities in the United States and Canada. As a member of a New York firm of consulting hydraulic and sanitary engineers for many years he served the Department of Water Supply, Gas and Electricity of that city.

Dr. Hering was the author of several technical books, and the recipient of the honorary degree of Doctor of Science from the University of Pennsylvania. He belonged to the American Society of Civil Engineers, the Canadian Society of Civil Engineers, the Institution of Civil Engineers, The Franklin Institute, the American Water Works Association, the New England Water Works Association and the American Public Health Association, and was a Fellow of the American Academy of Science. He became a member of the Society in 1906.

DONALD DAVIS HERR

Donald D. Herr, vice-president of the firm of Arthur G. McKee & Co., Cleveland, Ohio, died on September 1 in Miyanoshita, Japan, from injuries received during the violent earthquake on that date. Mr. Herr was born in March, 1880, in Bennington, Kansas. He was graduated from Pennsylvania State College in 1902 with the degree of B. S.

Upon graduation he was employed as draftsman with the American Steel & Wire Co., Cleveland, Ohio, but was soon advanced to the position of construction engineer. In 1905 he became superintendent of construction at the Clairton, Pa., plant of the Carnegie Steel Co., where he remained for about two years when he entered the employ of Arthur G. McKee, consulting and contracting engineer in Cleveland, to take personal charge of the reconstruction of the furnace of the St. Louis Blast Furnace Co., and of the first blast-furnace unit of the Inland Steel Co., Indiana Harbor, Ind. He was then placed in general charge of all construction work until the reorganization of the company when he became vice-president in charge of the field and sales departments.

Mr. Herr became a member of the Society in 1914. He did particularly noteworthy work during the War in directing construction work which included the removal of a complete blast-furnace plant from Midland, Ont., to the Soo plant of the Algoma Steel Co., completely rebuilding this plant under the most difficult conditions. He also had charge of a considerable part of the nitrate-plant equipment near Muscle Shoals.

JOSEPH S. HIBBS

Joseph S. Hibbs, for the past thirty years consulting engineer in foundry construction and practice and assistant general manager of the J. W. Paxson Co., Philadelphia, Pa., died on December 8, 1922. Mr. Hibbs was born on September 17, 1844, in Philadelphia. He was educated in the grammar and high schools of that city and later studied at the Franklin Institute.

From 1861 to 1865, during the Civil War, Mr. Hibbs served with the U. S. Army and at the close of the war started his apprenticeship as a molder. In 1869 he became connected with the Keystone Coöperative Manufacturing Co., as general manager. Ten years later he resigned from that position to become mechanical superintendent of H. R. Ives & Co.,

Montreal, Canada. In 1881 he became the traveling member of the firm of S. Obermeyer Co., where his duties consisted of selling and installing all kinds of foundry equipment. He was with this firm for eleven years when he left to take the position of assistant general manager with the J. W. Paxson Co. As a consulting engineer in foundry construction and practice, in which field he was an authority of wide reputation, he was active not only throughout the United States but also in Canada.

Mr. Hibbs became a member of the Society in 1922. He was also a member of the American Institute of Mining and Metallurgical Engineers, the American Ceramic Society, the American Academy of Political and Social Science, the Academy of Fine Arts, the masonic order, and of a number of socal clubs and church organizations.

H. JOHN O. HINCHEY

H. John O. Hinchey, who became a member of the Society in 1916, died on June 17, 1923. Mr. Hinchey was born in August, 1880, in Lansing, Mich., where he was educated.

He was first employed in 1889 by the Southern Exhaust & Blow Pipe Co., New Orleans, La., on erection and mechanical work. Four years later that firm was reorganized and the name changed to the National Blow Pipe Co. Mr. Hinchey remained as designer with the new firm. In 1906 he became connected with the Standard Blower Co., Atlanta, Ga., where his work dealt with designing and contracting. Six years later he resigned from that firm to take a position of a similar nature with the South Atlantic Blow Pipe Co., Savannah, Ga.

From 1914 to 1919 Mr. Hinchey was manager of the Atlanta office of the Buffalo Forge Co. and the Buffalo Steam Pump Co., when he left to become associated with the Miller Manufacturing Co., Chicago, Ill., with which concern he was associated at the time of his death.

NORMAN A. HOBSTETTER

Norman A. Hobstetter, sales engineer with the Lathrop-Hoge Construction Co., Cincinnati, Ohio, died on September 27, 1923. Mr. Hobstetter was born on September 22, 1893, in Pomeroy, Ohio, where he received his early education. For a number of years he taught school in his home district and then in 1915 entered the University of Cincinnati.

During the War he served in the U. S. Army and upon his discharge returned to the University and was graduated in 1920 with the degree of M.E. He served his apprenticeship, as part of his college course, with the Platt Iron Works, Dayton, Ohio. In 1920 he entered the employ of the Lathrop-Hoge Construction Co. Mr. Hobstetter became a junior member of the Society in 1920. He belonged to the University Club of Dayton and was a member of the masonic order.

GEORGE HORNUNG

George Hornung, widely known consulting engineer of Cincinnati, Ohio, died on November 27, 1923. Mr. Hornung was born on May 4, 1840, in Cincinnati, where he was educated and served his apprenticeship as a machinist. At the opening of the Civil War he enlisted as a private in the first Kentucky Volunteer Infantry. He was soon given a first lieutenant's commission and rendered distinguished service under General Rosecrans.

Upon return to civil life he became superintendent of the Newport, Ky., waterworks, where he remained four years when he became associated with Gray, Bowman & Co., St. Louis, builders and owners of water and gas works. Upon completing his work with that firm he opened his own office as consulting hydraulic and mechanical engineer with headquarters in

Cincinnati, in which field he was actively interested until the time of his death.

Mr. Hornung became a member of the Society in 1886. He belonged also to the American Water Works Association and to the Central States Water Works Association. He was a member of the Knights of Pythias.

T. W. HUGO

T. W. Hugo, a member of the Society since 1882 and one of the best known consulting engineers in Minnesota, died on February 27, 1923. Mr. Hugo was born on July 29, 1848, in Boddinac, Cornwall, England. His parents moved to Kingston, Ontario, Canada, when he was two years old and it was there that he was educated. Upon leaving school he served for five years as a machine apprentice in the Kingston Foundry & Engine Works, and then was employed for three years as second engineer and nine years as chief engineer on lake steamers. He superintended the cutting of the steamer *Campana* in two at Montreal, towing the two parts up the river and fastening them together again for Lake navigation.

In 1881 Mr. Hugo settled in Duluth, Minn., becoming chief engineer of Elevator " B " and continuing as chief engineer of the largest elevator plants in that city. He served as consulting engineer to many firms throughout the State and was recognized as a leader in that field.

Mr. Hugo was elected alderman of the fourth ward of Duluth in 1890 and on taking office was made president of the City Council, serving in that capacity for eight years. In 1900 he was elected mayor of Duluth and for four years was the city's chief executive. Since that time he was active on the school board, acting as president for many years.

He was one of the earliest members of our Society. He was also a member of the National Association of Stationary Engineers. Mr. Hugo belonged to the masonic order in which he had attained to one of the highest seats of masonry in the world.

GEORGE H. HULETT

George H. Hulett, one of America's most noteworthy inventors and a pioneer in the development of machinery for handling bulk freight, died in Daytona, Fla., on January 17, 1923. Mr. Hulett was born in Conneaut, Ohio, on September 26, 1847, and was educated in the Military School of Cleveland, Ohio.

In 1890 he became engaged in the manufacture of coal and ore-handling machinery in Cleveland and in 1898 became associated with the Variety Iron Works as construction engineer. Five years later he became engineer of the present McMyler Interstate Co. In 1907 he became connected with the Webster, Camp & Lane Co., Akron, which shortly afterward was consolidated with the Wellman-Seaver-Morgan Co., Cleveland. He was vice-president and director of this company until 1918 when he became vice-president of the Hulett Engineering Co., Cleveland.

Mr. Hulett's most distinguished engineering achievement was his invention of the automatic ore-unloading machine bearing his name. The first unloader of this type was erected for the Carnegie Steel Co., Conneaut, Ohio. It had a ten-ton bucket and was hydraulically operated. Many of the Hulett type machines have since been installed at ports on the Great Lakes and these have contributed greatly to reducing the cost of handling ore and to the speed of unloading boats.

Mr. Hulett became a member of the Society in 1903. He also belonged to the Cleveland Engineering Society, the Cleveland Chamber of Commerce and the Daytona Chamber of Commerce. He was a member of the masonic order.

JAMES LEON HUNT

James Leon Hunt was born in March, 1886, in Hartford, Conn., where he received his early education. He was graduated from New York University in 1913 with the degrees of B.S. and M.E. He had previously served an apprenticeship as machinist and toolmaker with the Sullivan Engine Co., the Arthur Gear Works, the H. R. Worthington Pump Co., and the DeLaval Steam Turbine Co.

In 1913 he became associated with the Alberger Pump & Condenser Co., New York, as testing and erecting engineer on pumps, turbines, condensers, etc. Three years later he accepted the position of assistant chief safety engineer with the Ocean Accident and Guarantee Corporation, New York City, where he remained a year when he left to become assistant to the superintendent of construction of the General Chemical Co. In 1918 Mr. Hunt became mechanical engineer in the operating department of the Ætna Explosives Co., resigning from that position to accept a commission as first lieutenant in the engineering division, trench-warfare section of the Ordnance Department of the U. S. Army.

He was associated with New York University from 1920 until the time of his death, in the capacity of supervising engineer of the erection of the new Sage Laboratory at University Heights. He was largely responsible for the detailed design of the installation of equipment in this building, which installation has proved one of the most up-to-date and unique in respect to its arrangement for experimental laboratory work.

For the year previous to the time of his death, his services were used to work up an operating plant for one of the University buildings at Washington Square.

Mr. Hunt became a junior member of the Society in 1914 and was advanced to the grade of associate member in 1919. He died on October 10, 1923.

ROBERT WOOLSTON HUNT

Robert Woolston Hunt, who less than a month before his death was presented the high honor of the Washington Award "for his pioneer work in the development of the steel industry in the United States and for a life devoted to the advancement of the engineering profession," died at his home in Chicago on July 11, 1923.

Captain Hunt was born December 9, 1838, in Fallsington, Bucks County, Pa. His father, Dr. Robert A. Hunt, died when he was only sixteen years of age, and in 1857 the family moved to Pottsville, Pa., where Robert spent several years in the iron rolling mill of John Burnish & Co., acquiring a practical knowledge of puddling, heating, rolling, and other details of the iron-rail business. Later he took a course in analytical inorganic chemistry in a laboratory in Philadelphia, upon the completion of which he entered the employ of the Cambria Iron Works at Johnstown, Pa. The first analytical laboratory maintained in America by an iron and steel company was established under his direction in 1860 for the Cambria Works.

In the Spring of 1861, as night foreman of the Elmira rolling mill, Elmira, N. Y., Captain Hunt assisted in the organization of the working force and the starting of that industry. In the Fall of that year he entered the U.S. Army, and was stationed at Harrisburg. In 1862 he was put in command of Camp Curtin, at Harrisburg, with the rank of captain. The next year he served the state of Pennsylvania as mustering officer, and in 1864 assisted in recruiting Lambert's Independent Mounted Company of Pennsylvania Volunteers.

After the close of the war Captain Hunt returned to the employ of the Cambria Iron Company and was sent to the experimental Bessemer works at Wyandotte, Mich., owned jointly by the Cambria Company and the

Kelly Pneumatic Process Company. The Kelly plant had in 1864 produced from pig iron remelted in a reverberatory the first penumatic steel made in America, and Captain Hunt, while in charge of the works there, introduced the use of a cupola for remelting. In 1866 he returned to Johnstown, where he had charge of the hammering and rolling of steel and superintended the rolling of the first steel rails made in America on a commercial order, produced for the Pennsylvania Railroad from ingots cast by the Pennsylvania Steel Company's plant at Steelton, Pa. The behavior of these ingots led to the successful introduction of blooming by rolling instead of hammering.

The Bessemer plant of the Cambria Company, finished in 1871, was designed and built by John Fritz, chief engineer of the company, with the assistance of A. L. Holley and Captain Hunt. It embodies several new features, such as the continuous running of the cupola, and a new system of bottom casting. Captain Hunt was in charge of these works from the time of their completion until August, 1873, when he resigned to become superintendent of the Bessemer Works of John A. Griswold & Co., at Troy, N. Y. These works were enlarged and remodelled, and the name of the company changed, first to the Albany & Rensselaer Steel & Iron Co., and later to the Troy Iron & Steel Co. Captain Hunt remained in charge until 1888, and during these years he supervised the erection of three large blast furnaces, and made many grades of Bessemer steel not previously produced in America, notably soft steel for drop forgings. He was also a pioneer in the production of steel for gun barrels, carriage axles, drills, and springs. He took out several patents on steel and iron metallurgical processes and machinery, and the Hunt-Jones-Suppes rail-mill feed tables developed during this period, were used later under licenses by the majority of the rail mills in the United States.

In 1888 Captain Hunt resigned his position at Troy and went to Chicago, where he established the firm of Robert W. Hunt & Co., consulting engineers, an organization which now has offices and laboratories in many cities.

The Washington Award is one of many tributes to the work of Captain Hunt. In 1912 he was awarded the John Fritz Medal, "for his contributions to the early development of the Bessemer Process," and in 1920 the Hunt Medal "for meritorious contributions to the art of making steel." He was for many years a trustee of the Rensselaer Polytechnic Institute, which in 1916 conferred upon him the honorary degree of Doctor of Engineering. While residing in Troy he was elected for four successive terms Commander of John A. Griswold Post, No. 338, G.A.R.

In the development of the engineering societies he has played an important part. He was president of the A.I.M.E. in 1883 and again in 1906; president of the A.S.M.E. in 1891; president of the Western Society of Engineers in 1893; president of the American Society for Testing Materials in 1912; and American Vice-President of the International Association for Testing Materials in 1914. He belonged also to the A.S.C.E., the Institution of Civil Engineers, the Institution of Mechanical Engineers, the Canadian Society of Civil Engineers, and the British Iron and Steel Institute. Captain Hunt was secretary of the Committee on Standard Rail Sections appointed by the A.S.C.E., the final report of which was made in 1893, and of the Special Committee on Rail Sections, appointed in 1892, which reported finally in 1910. The sections recommended by this committee were generally adopted by the railways of America.

His services as an officer and on committees of these societies, as well as his contributions to their proceedings, and lectures before educational and scientific bodies, have been of inestimable value. He was an honorary member of the A.I.M.E., A.S.M.E., American Society for Testing Materials, and Western Society of Engineers.

ARIYA INOKUTY

Ariya Inokuty, professor of mechanical engineering in the Imperial University of Tokyo, Japan, died on March 25, 1923, at the age of fifty-eight. Professor Inokuty was born in Ishikawa, Japan. He received the degrees of mechanical engineer and doctor of engineering from the College of Engineering of the Imperial University of Tokyo, and then for three years served as an instructor in mechanical engineering in the Japanese Navy.

Since 1886 Professor Inokuty occupied the chair of mechanical engineering in the Imperial University as professor in charge of that department. He designed and superintended the construction and installation of many water-power plants and pumping machinery of considerable size.

Professor Inokuty wrote a great number of papers on scientific and engineering subjects which have been collected and printed in one book. He became a member of the Society in 1915.

ABBOTT L. JOHNSON

Abbott L. Johnson, president of the Warner Gear Co. and of the Glascock Brothers Manufacturing Co., both of Muncie, Ohio, died at his home in Daytona, Fla., on January 11, 1923. Mr. Johnson was born in Herkimer County, N. Y., in August, 1852. When he was twelve years old his parents moved to Ashtabula, Ohio, where he received his education in the public schools.

There when only a few years out of school he established a bentwood factory. Later, in Bluffton, Ind., he founded another bentwood works and in a few years engaged in the hardwood lumber business at Montpelier, Ind., with J. T. Arnold, under the firm name of A. L. Johnson & Co. In 1880 he established a hardwood lumber mill in Muncie, Ind., and four years later formed a partnership with William E. Hitchcock, founding a factory for the manufacture of skewers and other hardwood products under the name of the Muncie Skewer Co.

At the time of his death, in addition to being president of the Warner Gear Co., Muncie, Ind., and the Glascock Brothers Manufacturing Co., he was director of the Delaware County National Bank and also of the Morgan-Hitchcock Co.

Mr. Johnson became a member of the Society in 1910. He belonged to a number of social clubs and organizations and was a member of the masonic order.

MAURICE LANDIN

Maurice Landin, who until September, 1922, was treasurer of the Boreal Chemical Products Corporation, New York, N. Y., died on March 26, 1923. Mr. Landin was born on November 30, 1893, in Koretz, Russia. He received his technical education in Cooper Union and later attended the evening sessions of Brooklyn Polytechnic Institute, receiving the degree of M.E. in 1919.

From 1913 to 1915 Mr. Landin was employed by the Grand Electric Co. on general engineering work. The following year he became connected with the Baldwin Locomotive Works, Philadelphia, on the detailing of locomotive parts and boiler design. In 1917 he resigned to enter the employ of the J. B. Arnold Construction Co., New York, where his work dealt with the appraising of railroad property. From 1918 to 1920 Mr. Landin was with the Public Service Commission of New York as junior electrical engineer in their department of equipment inspection. He left this position to organize the Boreal Chemical Products Corporation to manufacture dyes. He severed his connection as treasurer of this firm in 1922, intending to return to his own field but his illness prevented such action.

Mr. Landin became a junior member of the Society in 1919. He belonged also to the Zionist Engineers of America.

ANTOINE JEAN LANGELIER

Antoine Jean Langelier, president and designing engineer of the Langelier Manufacturing Co., Arlington and Cranston, R. I., died suddenly on September 18, 1923. Mr. Langelier was born on August 29, 1859, in St. Hyacinthe, Quebec, Canada, where he was educated.

While still a boy he came to the United States with his parents and took up the trade of machinist, serving a three-year apprenticeship with the Manchester Locomotive Works, Manchester, N. H. For seven years he was in the employ of the Brown & Sharpe Manufacturing Co., Providence, R. I., and then spent one year designing and building special machinery for the H. B. Smith Co., Westfield, Mass.

In 1887 with his father he established the Langelier Manufacturing Co., in Cranston, R. I. In 1917 the company incorporated and built a new plant at Arlington, which has been engaged in the manufacture of special machinery designed by Mr. Langelier. He had devoted much of his effort to hot and cold swaging of all kinds of metal. Mr. Langelier became a member of the Society in 1914.

CHARLES A. LINDSTROM

Charles A. Lindstrom, assistant to the president of the Pressed Steel Car Co., Pittsburgh, Pa., and a member of the Society since 1891, died in 1921. Mr. Lindstrom was born in 1853 in Sweden. He attended a technical school in Stockholm and then served an apprenticeship as a machinist.

He came to this country in charge of the Swedish exhibit of machinery at the Philadelphia Exposition in 1876. The following year he became foreman of the Altoona shops of the Pennsylvania Railroad. Two years later he became a draftsman in the shops and in 1881 chief draftsman. Later he became mechanical engineer in the Altoona office of the road.

About 1904 Mr. Lindstrom became associated with the Pressed Steel Car Co., Pittsburgh, Pa., as chief engineer, a position he held until 1914 when he became assistant to the president of the company.

ALBERT KING MANSFIELD

Albert King Mansfield, a member of the Society since 1887 and formerly mechanical engineer and a director of the Buckeye Engine Co., Salem, Ohio, died on January 23, 1923. Mr. Mansfield was born on March 25, 1849, in Newburyport, Mass. He received his early education in the schools of Lowell, Mass., and then worked for a year in the Lowell Machine Shops before entering Massachusetts Institute of Technology. He was graduated in 1873 and then went to Munich, Germany, to complete his studies in mechanical engineering.

Upon his return to this country he engaged for a short period in consulting mechanical and naval engineering work and then went to South America where for about three years he served as assistant in the astronomical observatory of the Argentine Republic at Cordova. Upon his return to this country he became connected with the Pan Handle Railway as master builder and was located in Steubenville, Ohio. Three years later he became superintendent of motive power of the New York & New England Railroad. His next position was as assistant manager of the Pullman Shops in Pullman, Ill. In 1884 he opened an office in Chicago and later in New York where he engaged in consulting work until 1888 when he accepted the position of mechanical engineer of the Buckeye Engine Co., Salem, Ohio. He remained with this company until 1902 when he resigned from the directorship to retire from business because of ill health.

During his connection with the Pan Handle Railway, Mr. Mansfield developed and patented several railroad devices, among which were the Mansfield water column and the Mansfield switch stand, both of which are

in general use today. Besides being a member of our Society he was also a member of the Archaeological Society of Washington, D. C., the National Geographic Society and the Society for the Advancement of Science.

WALDO H. MARSHALL

Waldo H. Marshall, chairman of the board of directors and president of the Consolidated Machine Tool Corporation of America, and former president of the American Locomotive Company, died August 23, 1923, at his summer home at Barnstable, Mass., after a brief illness.

Mr. Marshall was born in East Boston on June 7, 1864. He entered the railway field as draftsman for the Rhode Island Locomotive Works. From 1888 to 1897 he was engaged in editorial work on the *Railway Review, Railway Master Mechanic,* and *American Engineer and Railroad Journal.* For the next two years he was assistant superintendent of motive power for the Chicago & Northwestern Railroad.

In 1899 Mr. Marshall became associated with the Lake Shore & Michigan Southern Railway, serving it successively as superintendent of motive power, general superintendent of that road together with the Lake Erie & Western, and Indiana, Illinois & Iowa, and as general manager. He resigned these connections in 1906 to become president of the American Locomotive Company, a position which he occupied until 1917.

For a time during 1917 he was affiliated with J. P. Morgan & Co. but left that firm in January, 1918, to accept the appointment of chief of the production division of the Ordnance Department, U. S. Army. He served as one of the dollar-a-year men throughout the War, being a member of the Naval Consulting Board, and of the Committee on Industrial Preparedness of New York.

Following the War Mr. Marshall was associated with T. A. Gillespie Company and the Gillespie Manufacturing Company, being chairman of the board of the latter company. In March, 1923, upon the death of C. K. Lassiter, who was president of the Consolidated Machine Tool Corporation and whom he helped in organizing the corporation as a merger of five companies, Mr. Marshall, already serving as chairman of the board of directors, also took over the duties of president.

Mr. Marshall was also a director of the American Brake Shoe & Foundry Company, the Bucyrus Company, and the Chatham and Phenix Bank of New York City. He became a member of the Society in 1893 and served on its Finance Committee in 1912–1914. He was chairman of the Executive Committee of the Ordnance Division at the time of his death.

FREDERICK SHELDON MARTIN

Frederick S. Martin, superintendent with the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa., died on April 1, 1923. Mr. Martin was born on February 12, 1863, in Claverack, N. Y. He received his early education in private schools of Poughkeepsie, N. Y., and then entered the Stanstead Wesleyan College, Stanstead, Quebec, Canada, from which he received a commercial degree in 1878.

Mr. Martin served his apprenticeship with the Franklin Foundry & Machine Co., Providence, R. I., and then was employed by the Brown & Sharpe Manufacturing Co., with charge of the planing and milling department in Shop No. 2. Later he became connected with the Edison General Electric Co., Schenectady, N. Y., where for three years he served as foreman of the gear and milling departments. For a number of years Mr. Martin was superintendent of works for the R. D. Nuttall Co., Pittsburgh, Pa., resigning from that position to become associated with the Westinghouse Electric & Manufacturing Co.

Mr. Martin became a member of the Society in 1905. He belonged to

the Veteran Employes' Association, the Foremen's Association and the Superintendents' Council of the Westinghouse Co.

JOHN FRANCIS MARTIN

John Francis Martin, who, until his retirement because of ill health, was superintendent of the Neptune Meter Co., Long Island City, N. Y., died on June 26, 1923. Mr. Martin was born on July 27, 1868, in Hartford, Conn., where he was educated. He served a four-year apprenticeship with the Dwight-Slate Machine Co., Hartford, and then became connected with the Pratt-Whitney Co., also of Hartford.

In 1894 Mr. Martin became assistant foreman of the tool room and machine shop of the Pope Manufacturing Co., Hartford, designers and builders of tools and fixtures for producing bicycle parts. He was with this concern for nine years when he resigned to take charge of the experimental work of the John Underwood Adding Machine Co. with offices in Hartford and New York City. Eight years later he became associated with the Neptune Meter Co., as assistant superintendent and later as superintendent, where he had charge of designing and building tools, fixtures and machines for producing water-meter and ammunition parts.

Mr. Martin became an associate member of the Society in 1918.

JOHN ALEXANDER McCAMPBELL

John A. McCampbell, superintendent of construction for the Stone & Webster Corporation, Boston, Mass., died on May 24, 1923. Mr. McCampbell was born on April 3, 1873, in Knoxville, Tenn., where he was educated. He served an apprenticeship as a machinist with the Knoxville Foundry & Machine Co., and then became a journeyman wireman in Cramp's Shipyard in Philadelphia, Pa.

In 1894 he became connected with the Pepper & Register Co., Philadelphia, engineers and contractors, and he was with this firm until 1902, having had charge of the track, overhead-line and underground construction and power station. In 1902 Mr. McCampbell became chief operating engineer for the Houghton Co., but returned after a year to the Pepper & Register Co. as superintendent of construction of high-tension, transmission lines, track and power station. From 1905 to 1907, the last two years of his connection with this firm, he was master mechanic of the I. & I. Railway, Clinton, Ohio. Since 1907 Mr. McCampbell was associated with the Stone & Webster Coporation, as superintendent of construction of high-tension transmission, trolleys, underground and power stations.

Mr. McCampbell became a member of the Society in 1919. He belonged to the masonic order and was a Shriner.

H. S. McDEWELL

H. S. McDewell, research engineer of the Maxwell Motor Co., Detroit, Mich., died suddenly on April 6, 1923, at his home in Washington, D. C. Mr. McDewell was born on July 9, 1885, in Winthrop, Mass. He was prepared for college at the Phillips Exeter Academy and then entered Harvard University from which he was graduated in 1903 with the degree of M.E. In 1908 he received the degree of M.M.E. from the Graduate School of Applied Science of Harvard.

In that same year he became associated with the Allis-Chalmers Manufacturing Co., Milwaukee, Wis., as gas-engine erecting engineer. He was with that company until 1914 when he resigned to become an instructor in internal-combustion engineering and laboratory practice at the University of Illinois, where he was located until 1917. At that time he became engineer of tests in the aeronautical engine-testing laboratory at the Washington

Navy Yard, supervising the testing and development of aeronautical engines and power-plant accessories for a period of about five years. Subsequently he was associated with the Maxwell Motor Co.

Mr. McDewell collaborated with Prof. L. S. Marks in writing Gas and Oil Engines and Gas Producers and in preparing the manuscript for the internal-combustion section of the Mechanical Engineer's Handbook. He became a junior member of our Society in 1908. He was also a member of the Society of Automotive Engineers and a charter life member of the Association of Harvard Engineers.

JOHN B. McGOWAN

John B. McGowan, operating superintendent of the Cunard Building, New York, N. Y., died on January 31, 1923. Mr. McGowan was born on February 7, 1872, in New York, N. Y. He was educated in New York schools and attended St. Francis Xavier College.

From 1893 to 1918 he was connected with George Borgfeldt & Co., New York City, as mechanical engineer and superintendent of construction. In this capacity he designed and supervised the mechanical equipment of their properties. He was also in charge of redesigning and reconstructing the mechanical novelties of which this firm was a large importer.

He was associated for a short period in 1919 with the Consolidated Card Co., Long Island City, N. Y., as superintendent of construction. He resigned from this position to become operating superintendent of the Cunard Building.

Mr. McGowan became a member of the Society in 1922. He also belonged to the Geographical Society and to the National Association of Stationary Engineers.

FORRES McGRAW

Forres McGraw, secretary and treasurer of the Swanson-McGraw Co., Inc., New Orleans, La., died on September 21, 1923. Mr. McGraw was born on October 25, 1895, in New Orleans, La., where he received his early education. He was graduated in 1917 from the University of Tulane with the degree of B.S. in mechanical engineering.

During the War Mr. McGraw served as a captain of Field Artillery in the U.S. Army and upon being honorably discharged at its close, entered the employ of Fairbanks-Morse & Co., in their engineering department. Later he became secretary and treasurer of the Swanson-McGraw Co., Inc., which position he held at the time of his death.

Mr. McGraw became a junior member of the Society in 1921.

JOSEPH H. McNEILL

Joseph H. McNeill, chief inspector of the Hartford Steam Boiler & Insurance Co., died suddenly on April 18, 1923, of heart failure. Mr. McNeill was born on February 14, 1865, in Charlottetown, Prince Edward Island, where he received his education in the public schools and the Prince of Wales College.

In his early life he was connected with the Prince Edward Island Railway. He served also as chief engineer of ocean-going vessels under licenses both from the United States Government and from the British Board of Trade and for several years was an engineer officer of the Eastern Steamship Co., sailing out of Boston, Mass. Later he held the position of engineer with the New England Piano Co., Cambridge, Mass., and of the Hood Rubber Co., Watertown, Mass. He resigned from the latter position in 1889 to enter the service of the State of Massachusetts as a boiler inspector of the District Police of the Commonwealth of Massachusetts. About ten years later he became chief inspector and chairman of the Massachusetts

Board of Boiler Rules. In both capacities he had much influence in shaping the laws and rules governing the construction, installation and operation of boilers in that state.

In 1912 Mr. McNeill resigned from that position to become chief inspector of the Boston department of the Hartford Steam Boiler Inspection & Insurance Co. He remained in Boston until 1915 when he was transferred to the New York office of the company where, as chief inspector, he continued until his death to have charge of inspection work in the State of New York and sections of Vermont and New Jersey.

Mr. McNeill became a member of the Society in 1911. He was a member of the Board of Examiners of the Industrial Commission, Boiler Inspection Department, State of New York. He belonged to the masonic order.

HARVEY MIDDLETON

Harvey Middleton, a member of the Society since 1886, died on March 30, 1923. Mr. Middleton was born on June 27, 1852, in Philadelphia, Pa., where he was educated. Upon being graduated from high school he served a four-year apprenticeship in the machine shops of Wm. Sellers & Co., Philadelphia, and then entered the shops of the Pennsylvania Railroad at Renova, Pa. He rose through the various positions to that of master mechanic.

Following his shop work in Renova and through positions in the St. Paul & Manitoba Railroad, he was made superintendent of motive power of the Louisville & Nashville Railroad, with headquarters at Louisville, Ky. After several years with this road he resigned to accept a similar position with the Santa Fe Railroad with headquarters at Topeka, Kan. From 1890 to 1896 Mr. Middleton was manager of the Pullman Co., at Pullman, Ill.

In 1896 he accepted the position of general superintendent of motive power of the Baltimore & Ohio Railroad at Baltimore, Md., remaining in that capacity until 1900 when he became president of the Thomas C. Basshor Co., engineers and contractors of Baltimore. He held this position until the dissolution of the company in 1914.

Although retired at the outbreak of the War, Mr. Middleton resumed his activities by becoming connected with the Ordnance Department of the U. S. Army as consulting engineer having to do with production and contracts and finally the various post-war settlements. Upon the completion of this work he again retired, living in Baltimore until his death.

Besides belonging to our Society, Mr. Middleton was a member of the Engineers' Club of New York, the Maryland Club of Baltimore and the American Society of Colonial Wars.

SAMUEL L. MOORE

Samuel L. Moore, vice-president and general manager of the Moore Brothers Co., Elizabeth, N. J., died on March 17, 1923. Mr. Moore was born in 1852 in Elizabeth, N. J. From 1869 to 1873 he served his apprenticeship as machinist with S. L. Moore & Son. In 1873 he qualified for a marine engineer's license and for the next six years followed the sea.

In 1879 Mr. Moore became foreman of the machine shop of S. L. Moore & Sons Co., a position which he held until 1887 when he became president and general manager of the Moore Brothers Co. Mr. Moore became a member of the Society in 1907.

LEWIS HALLOCK NASH

Lewis H. Nash, president of the Nash Engineering Co., South Norwalk, Conn., died on November 17, 1923. Mr. Nash was born on April 17, 1852,

in Norwalk, Conn., where he received his early education. He learned his trade of a first-class machinist in the Norwalk Iron Works and then entered Stevens Institute of Technology in 1873, receiving in 1877 the degree of M.E.

Soon after being graduated he started work with the National Meter Co., of Brooklyn, N. Y., where eventually he became chief engineer. Nearly all of Mr. Nash's patents, which numbered over 400, were assigned to this concern until he left their employ in 1914. Among his patents were sixty or more on water meters and as many more on the design and operation of gas engines. In 1914 he established the Nash Engineering Co. of which he became president.

Mr. Nash became a member of the Society in 1891. In 1921 the degree of Doctor of Engineering was conferred upon him by his Alma Mater. In 1922 he was elected representative of the Connecticut State Assembly and sat in the 1923 session.

ROBERT S. OBERLY

Robert S. Oberly, Major in the Ordnance Department of the U. S. Army and chief of the Arsenal and Industrial Relations Section, Office of the Chief of Ordnance, Washington, D. C., died on September 4, 1923. Major Oberly was born on September 26, 1885, in Easton, Pa. He was educated in the public schools of Pennsylvania and was graduated from Cornell University in 1908 with the degree of M.E. Prior to entering Cornell he had been employed in several manufacturing establishments as a machinist.

Major Oberly entered the U. S. Army as a second lieutenant in February, 1912, and during the World War held the rank of lieutenant colonel, Ordnance Department, being assigned to duty as executive assistant to the commanding officer of the Aberdeen Proving Ground. His work in that position involved concrete construction, road work, building construction and design, machinery layout, gun-emplacement construction and design, velocity apparatus, power-plant layout, etc.

Major Oberly became a member of the Society in 1919. He belonged also to the American Society for Testing Materials, the Army Ordnance Association and the masonic fraternity.

THOMAS T. PARKER

Thomas T. Parker, who. at the time of his retirement in April, 1920, was chief boiler inspector and assistant superintendent of the boiler-insurance department of the Fidelity & Casualty Co., New York, N. Y., died at his home in Chicago, Ill., on March 28, 1923.

Mr. Parker was born on October 7, 1856, in Paris, Ky. He received his early education in Delaware, Ohio, and entered Wesleyan College there to study for the ministry. Because of his father's death, it became necessary for him to leave school at the age of thirteen. He became an apprentice machinist in a machine and boiler shop and worked through the stages of boiler maker, operating engineer, erector and indicator expert.

In 1891 he became a boiler inspector for the Fidelity & Casualty Co. His work carried him through many states until at length he settled in Chicago as chief boiler inspector for that firm. About twenty years ago he was called to the home office of the company to become assistant superintendent of the boiler-insurance department. He held this position until his retirement in 1920 because of ill health.

Mr. Parker was well known in the boiler world and was frequently called upon to serve in a consulting capacity. He designed and patented several improvements on steam boilers and also held patents on methods dealing with the elimination of scale-forming impurities. He became a member of the Society in 1914.

J. WILLIAM PETERSON

J. William Peterson, until recently president and owner of the Richardson-Phenix Co., Milwaukee, Wis., died suddenly on July 24, 1923. Mr. Peterson was born on December 16, 1875, in Morris, Ill. He finished his grammar-school education in Chicago and from that time on developed into a self-educated, self-made man. He entered the electrical trade in Chicago and gradually worked up into electrical contracting.

In 1903 he went to Cuba as a representative of the Westinghouse Electric & Manufacturing Co., returning in two years to accept a position with the American Blower Co. His next position was with the New York City office of the Hooven-Owens-Rentschler Co., where he remained until 1908 when he became connected with the Sight Feed Oil Co., Milwaukee, becoming its first New York agent. He was at the same time operating his own company known as the Peterson Engineering Co., specializing in large oiling-system installations.

In 1910 Mr. Peterson formed a partnership with the late W. E. Richardson, and taking over the Phenix Lubricator Co., Chicago, they created the Richardson-Phenix Co. In 1916 this company was merged with the Peterson Engineering Co. In January, 1922, Mr. Peterson retired from business.

Mr. Peterson became a member of the Society in 1911. He was also a member of the American Society of Refrigerating Engineers and the National Association of Stationary Engineers. He was one of Milwaukee's most prominent citizens, being associated with every large piece of civic work done in that city during the last decade. He was a member of the masonic fraternity.

CECIL O. PHILLIPS

Cecil O. Phillips, a director and vice-president in charge of manufacturing of the American Cotton Oil Co., and its subsidiaries, died on May 1, 1923. Mr. Phillips was born in May, 1866, in New York City, where he received his early education. Later he attended Stevens Institute of Technology for one year.

From 1886 to 1888 he served first as draftsman and then as designer with the St. Louis & Santa Fe Railroad. He then spent a year as chief draftsman with the Atlantic & Pacific Railway and then a year as resident engineer on construction with the Nevada & Fort Smith Railway. From 1891 to 1895 he served as superintendent of the Eagle Oil Works, Claremont, N. J. In 1895 he became associated with the American Cotton Oil Co., New York City, as mechanical engineer.

After a few years of general work in connection with the mechanical. department, he became identified primarily with the crushing mills of the organization and had charge of their operation. In 1914 he became secretary and treasurer of the Union Seed and Fertilizer Co., one of the subsidiary companies. Five years later Mr. Phillips was made vice-president of the American Cotton Oil Co. in charge of manufacturing and in this capacity had charge not only of the cotton-oil mills but of the refineries and soap plants. He was responsible for many improvements and advancements in the industry along mechanical operating lines and introduced into the plants many of the successful manufacturing processes which are now being used. At the time of his death he was a director of the company.

Mr. Phillips became a member of the Society in 1914.

RICHARD E. POWERS

Richard E. Powers, construction engineer for the Oneida Community, Ltd., Oneida, N. Y., died on September 26, 1923. Mr. Powers was born on May 6, 1877, in Hounsfield, N. Y. He was educated in the common and high schools of Watertown, N. Y., and learned the machinist's trade in the locomotive works of that town.

Mr. Powers then moved to Oneida, N. Y., and was employed by the Oneida Community, Ltd., as a mechanic and tool maker. He was soon made foreman of the machine shop and by study and energy advanced to the position of master mechanic. During the last few years he was construction engineer for the Community.

Mr. Powers became an associate member of the Society in 1917. He belonged also to the American Society of Heat Treating and was a member of the local order of Elks.

EDWARD OWNER QUINN

Edward O. Quinn, president and manager of the Blue Diamond Materials Co., West Roxbury, Mass., died on September 8, 1923. Mr. Quinn was born in June, 1882, in Los Angeles, Cal. He was graduated from Purdue University in 1904 with the degree of M.E.

He was first employed by the Coalingua Oil & Transportation Co., Coalingua, Cal., as an inspector on pipe-line construction and pumping plants. Later he was associated with the Doak Gas Engine Co., the Western Fuel Co., and the Western Calcium Co., all of San Francisco. From 1910 to 1918 he was connected for varying periods with the following concerns: the Round Rock Lime Co., Round Rock, Tex., the Western Lime & Plaster Co., Lime, Ore., the L. B. Hammond Machinery Co., Portland, Ore., the Cowell Portland Cement Co., Cowell, Cal., and the Henry Cowell Lime & Cement Co., San Francisco. From 1918 to 1920 Mr. Quinn was engaged in the construction of the Union Oil Refineries at Olium and Wilmington, Cal. The following year he designed and constructed the New England Oil Refinery, Fall River, Mass. In 1921 he became president and manager of the Blue Diamonds Materials Co., which position he held at the time of his death.

Mr. Quinn became a member of the Society in 1916. He belonged to a number of clubs and was a 32d degree Mason.

T. W. RANSOM

Tom Wells Ransom, consulting engineer of San Francisco, Cal., died on December 20, 1923. Mr. Ransom was born on October 11, 1869, in San Francisco, where he received his early education. He was graduated in 1891 from the University of California with the degree of B.S. For the next five years he was employed in the drawing office of the Union Iron Works and the Fulton Iron Works, also becoming familiar with machine-shop and foundry practice.

At the end of that time he became associated with John H. Hopps under the firm name of Hopps & Ransom, consulting engineers. The fire of 1906 ended this connection and Mr. Ransom continued alone in private practice. In 1911 he was engaged by the City of San Francisco to design and superintend the construction of a high-pressure fire system. Upon the completion of this piece of work, which is considered especially fine, he was retained by the City to design and build the Municipal Railway System, which he did successfully. In 1916 he resigned from the City Engineer's Office and resumed private practice until 1918 when he entered the service of the U. S. Shipping Board with headquarters in Los Angeles to supervise the wartime work of building and repairing vessels in that district. From 1919 to the time of his death Mr. Ranson had been engaged in private consulting work.

Mr. Ransom became a member of the Society in 1909. He was a member of the University Club of San Francisco.

ROGER P. REDIER

Roger P. Redier, formerly general sales manager of the Allied Machinery Co. of America, New York City, died in July, 1923. Mr. Redier was born in

May, 1878, in Paris, France, and always maintained his citizenship in that country. He was educated in French schools and attended the College de Dieppe.

Mr. Redier came to this country in 1897 and served a two-year apprenticeship with the Brown & Sharpe Manufacturing Co., Providence, R. I. He then returned to France and served in the French Army for four years. From 1904 to 1910 he was associated as field engineer with Alfred Herbert, Ltd., Coventry, England, one of the largest manufacturers of machine tools in England. At the end of that period he became general manager in charge of design and production of machine tools for Ste. Francaise de Machines Outils, Paris. Three years later he became general manager of R. S. Stockvis & Son, Paris. During the first year of the War Mr. Redier saw active service with the French Army. He was then sent to the United States to procure all necessary equipment for rifle and cartridge plants.

In 1919 Mr. Redier became connected with the Allied Machinery Co. of America as general sales manager. He resig. from this position in 1921 because of ill health. He became a member of the Society in 1919.

JOSEPH B. RIDER

Joseph B. Rider, consulting engineer, New York City, died on February 26, 1923. Mr. Rider was born on April 28, 1868, in Norwalk, Conn., where he received his early education. Later he attended Rensselaer Polytechnic Institute, being graduated in 1889 with the degree of C.E. The following year he took post graduate work in Lafayette College.

During his early professional career he served as associate editor of *Fire and Water* and contributed to the technical press. He was also the author of Rider's Little Engineer.

Mr. Rider became associated with his father in 1904 in designing and supervising hydraulic and water-power work, making a specialty of municipal water-works plants. Upon his father's death, Mr. Rider carried on the business and enlarged its scope to include electric-light, power-plant and hydroelectric work. He acted as expert in numerous valuation and condemnation proceedings.

During recent years Mr. Rider devoted a great deal of time to consulting work for public and quasi-public works and utilities and was identified with a large number of such works throughout the country.

Mr. Rider became a member of the Society in 1916. He was also a member of the American Water Works Association and the American Association of Engineers.

JOHN I. ROGERS

John I. Rogers, founder, treasurer and engineer of the Eagan-Rogers Steel & Iron Co., Crum Lynne, Pa., died on August 15, 1923. Mr. Rogers was born on September 15, 1882, in Philadelphia, Pa., where he was educated. He was graduated from the University of Pennsylvania in 1904.

From 1904 to 1909 he was in the employ of the Midvale Steel Co., where he worked through the shops to the superintendency of the rolled-steel-wheel plant. At the end of that time he resigned to enter the consulting field, making a specialty of steel and forging plants. He designed plants for the Bridgeport Forge Co., for the Heffenstahl Forge & Knife Co., and for many others. In 1912 together with Daniel C. Eagan, he organized the Eagan-Rogers Steel & Iron Co. and built their present plant for the manufacture of steel castings.

Mr. Rogers became a junior member of the Society in 1910 and in 1921 was advanced to full membership. He belonged also to the American Society for Testing Materials, the Engineers' Club of Philadelphia, the Merion Post of the American Legion and to several social clubs.

During the War Mr. Rogers served as a lieutenant in the U. S. Navy, in command of the U.S.S. *Helmeta*, attached to the Atlantic Fleet.

ALEXANDER ROSE

Alexander Rose, mechanical engineer with J. E. Sirrine & Co., Greenville, S. C., died on July 11, 1923. Mr. Rose was born on February 20, 1867, in Charleston, S. C., where he received his early education. Later he attended Cornell University for three years.

From 1894 to 1906 Mr. Rose was associated with the following firms in the capacities noted: Morgan Iron Works, Spartanburg, S. C., master mechanic; Land Pebble Phosphate Co., Bartow, Fla., master mechanic; Palmetto Phosphate Co., Bartow, Fla., designer for phosphate mining plant; Charleston Mining Co., chief engineer of the Horseshoe Mines; Olympia Cotton Mills, Columbia, S. C., chief engineer; Darlington Cotton Mills, Darlington, S. C., and Laurens and Watts Mills, Laurens, S. C., chief engineer.

In 1906 Mr. Rose became connected with J. E. Sirrine & Co. as mechanical engineer in charge of the steam-power and heating departments, which position he held at the time of his death. He became a member of the Society in 1921.

JOHN HOWARD ROWEN

John Howard Rowen, commander, U. S. Navy, retired, and member of the faculty of the engineering college of the University of Minnesota, Minneapolis, Minn., died on September 7, 1923. Professor Rowen was born on January 25, 1871, in Philadelphia, Pa., where he received his early education. At the close of his third year in high school he was appointed to the Naval Academy at Annapolis where he was graduated from the engineering course in 1891.

He was commissioned as assistant engineer and during the next twenty years passed through the various naval grades to that of commander, retiring in 1911. For two years he was engaged in the study of archeology. In 1913 he became a member of the faculty of Toledo University, Toledo, Ohio, in charge of mechanical engineering. Two years later Professor Rowen accepted a position on the faculty of the University of Michigan, from which he was recalled to active service when the United States entered the World War. He served as naval inspector of machinery at the Allis-Chalmers Co., Milwaukee, Wis. In 1921 he became associated with the University of Minnesota and at the time of his death was associate professor of steam-power engineering.

Professor Rowen became a member of the Society in 1917. He was also a member of the American Association for the Advancement of Science, the National Geographic Society, the Society of Naval Architects and Marine Engineers, the Society for the Promotion of Engineering Education, the Engineers' Club of Minneapolis and of various naval clubs.

JOHN WARREN SARGENT

John Warren Sargent, consulting engineer of Providence, R. I., and widely known as an expert designer of stationary steam engines, died on January 31, 1923. Mr. Sargent was born on February 4, 1856, in South Amesbury, Mass., where he received his early education. Later he studied for two years as a special student in the Massachusetts Institute of Technology.

Previous to this course he had served an apprenticeship as a machinist with the Amesbury Machine Works. In 1881 he was employed as a machinist in William Cramp's Shipbuilding Works in Philadelphia, Pa. Later he was connected with the Dickson Manufacturing Co., Scranton, Pa., as draftsman and mechanical engineer.

In 1893 Mr. Sargent became consulting engineer to the Providence Steam Engine Co., Providence, R. I., and the following year entered into

partnership with a Mr. Rice in the firm of the Rice & Sargent Engine Co. Five years later the name of the firm was changed to the Providence Engineering Works and Mr. Sargent became its director and consulting engineer. He remained with this company until 1913 when he resigned to enter the consulting field of steam plants and engine regulation. Since 1920 Mr. Sargent had devoted his time to experimental research work, inventing and designing an inertia engine.

Mr. Sargent was one of the early members of the Society, joining the organization in 1886. He was a member of the Providence Engineering Society and an honorary member of the National Association of Stationary Engineers.

CHARLES ROBERTS SEABROOK

Charles Roberts Seabrook, associated with the firm of McClintock & Marshall, Pittsburgh, Pa., died on March 1, 1923. Mr. Seabrook was born on July 1, 1887, in Westminster, Md. He received his early education in the Shenandoah Valley Military Academy and then attended Newbury College, South Carolina, from which he was graduated in 1905 with the degree of B.S., later taking special work at the Armour Institute of Technology.

From 1905 to 1909 Mr. Seabrook worked on various engineering projects in southern states and Central America as draftsman, rodman, transitman and detailer. At the end of that period he became structural draftsman and detailer for the Virginia Bridge & Iron Co., Roanoke, Va. The following year he became squad foreman and assistant chief draftsman for the Cambria Steel Co., Johnstown, Pa. From 1912 to 1917 Mr. Seabrook was connected with the following firms: Eastern Steel Co., New York City, as designer and estimator; Heyl & Patterson, Inc., Pittsburgh, Pa., as squad foreman and designer; Lackawanna Bridge Co., Buffalo, N. Y., as squad foreman and assistant chief draftsman; Westinghouse, Church, Kerr & Co., New York City, as structural designer; Chile Exploration Co., Chuquicamata, Chile, as structural engineer and assistant superintendent of construction.

In 1918 Mr. Seabrook became civilian structural engineer in the Pittsburgh District of the Ordnance Department of the Army. His next position was as chief engineer of the H. M. Lane Co., Detroit, Mich. He was for a time consulting member of the firm of Owen & Seabrook, consulting engineers, Detroit, resigning to become associated with the firm of McClintock & Marshall.

Mr. Seabrook became a member of the Society in 1920. He was also a member of the Detroit Engineering Society, the Society of American Military Engineers and of several social clubs.

WILLARD THOMAS SEARS

Willard T. Sears, general mechanical engineer with the Niles-Bement-Pond Co., New York, N. Y., died on March 24, 1923. Mr. Sears was born on November 17, 1865, in Plymouth, Mass., where he attended the public schools. He was graduated from the Massachusetts Institute of Technology in 1887.

His first position was with the Holly Manufacturing Co., builders of pumps at Lockwood, N. Y., as draftsman. Later he was connected with the Pennsylvania Steel Co., Steelton, Pa., as a special designer on draw bridges and draw-bridge machinery.

For the past twenty-two years Mr. Sears was general mechanical engineer with the Niles-Bement-Pond Co., New York, N. Y., manufacturers of machine tools. During this period he was responsible for a number of their more important designs. In 1915 he represented the firm in China in a consultation with the government officials of that country on machinery for arsenals.

Mr. Sears became a member of the Society in 1889. He was also a member of the American Society for Testing Materials and of several social clubs. He belonged to the masonic order.

H. J. SMALL

H. J. Small, a life member of the Society since 1884 and vice-president from 1893 to 1895, died in the Fall of 1921. Mr. Small studied mechanical drawing and engineering at Mechanic's Institute in Toronto, Ontario, and then served a three-year apprenticeship as a pattern maker in the shops of the C. & N. W. Railway in Chicago and a similar period with the Kansas Pacific Railway as machinist and draftsman. He then became a draftsman in the employ of the Northern Pacific Railway.

Later Mr. Small held successively the positions of general foreman of the International & Great Northern Railway, master mechanic of the Galveston, Houston & Henderson Railway, general manager and master car builder of the Texas & Pacific Railway, and assistant superintendent of machinery of the Northern Pacific in charge of locomotive car shops. His last position and one which he held for many years before his retirement was as general superintendent of motive power of the Southern Pacific Railway.

CLARENCE LLEWELLYN SMITH

Clarence L. Smith, design engineer, with the Western Electric Co., Inc., New York City, died from accidental drowning on November 4, 1923. Mr. Smith was born on October 17, 1894, in Vinalhaven, Me., where he received his early education. He was graduated from the University of Maine in 1917 with the degree of B.S. in mechanical engineering, and then entered the student course of the Western Electric Co. in Chicago, Ill.

After a six months' training he was assigned to the manufacturing department where his work covered the development of a wire-drawing machine, an annealing furnace and machines for drilling diamond dies for wire drawing. In July, 1918, Mr. Smith was transferred to the New York office of the company where until the time of his death he was engaged in the development of central-office telephone and switchboard apparatus.

Mr. Smith became a junior member of the Society in 1923. He was a member of the honorary engineering fraternity Tau Beta Pi and also of Phi Kappa Phi.

JULIO FEDERICO SORZANO

Julio Federico Sorzano, consulting civil and mechanical engineer, New York City, died on June 25, 1923. Mr. Sorzano was born on October 25, 1852, in Cuba. He was educated in France and Belgium and was graduated in 1872 from the Institute de l'Etat Gembloux. He specialized in the sugar industry, and although established in New York, did the greater part of his work abroad, principally in Cuba, Mexico, Porto Rico, Haiti and the countries of South America.

Mr. Sorzano designed and built a score or more of sugar mills in Cuba and Porto Rico alone and was an acknowledged expert in matters pertaining to the raw-cane and beet-root sugar industries, in which he was one of the pioneers. He built the first railway in Venezuela, as well as two subsequent roads in the same country. He was consulted in connection with the foundations of the Brooklyn Bridge towers with which there had been considerable difficulty. He was greatly interested and very active in the Panama Canal controversy over the relative merits of the lock as against the sea-level canal, and was a member of the committee appointed by the Chamber of Commerce of New York to study the matter.

A close student of Latin-American affairs, he organized in 1911 the Pan-American Chamber of Commerce to foster friendly and trade relations and was president of that institution up to the time of his last illness. Mr. Sorzano was a life member of the Society, joining the organization in 1884. He belonged also to the American Society of Civil Engineers, the Institution of Civil Engineers of Great Britain, la Société des Ingénieurs Civils de France and the Chamber of Commerce of the State of New York.

CHARLES P. STEINMETZ

Charles Proteus Steinmetz, chief consulting engineer of the General Electric Company, died suddenly at his home in Schenectady, N. Y., October 26, 1923. He was born April 9, 1865, in Breslau, Germany, and was educated at the gymnasium (high school) and then at the University of Breslau, where he studied mathematics and astronomy, then physics and chemistry, and finally, for a short time, medicine and economics. Involved in the social democratic agitation against the government, he escaped to Switzerland in 1888, and there studied mechanical engineering at the Polytechnicum.

In 1889 he immigrated to America, and found a position with the Osterheld & Eickemeyer Manufacturing Company, first as draftsman, then as electrical engineer and designer, and finally in research work in charge of the Eickemeyer Laboratory, in New York.

With the absorption of the Eickemeyer-Field interests by the General Electric Company, Dr. Steinmetz joined the latter organization. Upon the transfer of the company's headquarters to Schenectady in the spring of 1894, Dr. Steinmetz organized and took charge of the calculation and design of the company's apparatus and of its research and development work.

For a number of years Dr. Steinmetz was professor of electrical engineering at Union College, and at the time of his demise was professor of electro-physics at Union University. During this period, however, he retained his connection with the General Electric Company as chief consulting engineer.

His incomparable ability for mathematical analysis found an avenue for utilization in investigations of magnetism, alternating currents, radiation, thermodynamics, and illumination, and his contributions made possible the present state of the art of generation and transmission of electrical power.

Dr. Steinmetz was past-president of the National Association of Corporation Schools, vice-president of the International Association of Municipal Electricians, past-president of the American Institute of Electrical Engineers, honorary member of the National Electric Light Association, past-president of the Illuminating Engineering Society, fellow of the American Association for the Advancement of Science, member of the (British) Institution of Electrical Engineers, member of The American Society of Mechanical Engineers, the Electrochemical Society, the Illuminating Engineering Society, the Physical Society, etc., etc. He had also served as president of the Common Council and twice as president of the Board of Education of the city of Schenectady.

THOMAS S. THOMSON

Thomas S. Thomson, mechanical superintendent of the Biltmore Hotel in Los Angeles, Cal., died on June 3, 1923. Mr. Thomson was born on March 3, 1867, in Waukesha, Wis. He received his early education in the schools of Chicago and attended the College of Pharmacy planning to go to the Rush Medical College but his health called for out-door work and he gained employment in the Union Stock Yards where he soon became yard master.

In 1894, however, he turned his attention elsewhere and after a corre-

spondence course in steam and electrical engineering, decided to go West. The following year he became engineer in the San Gabriel Hotel, San Gabriel, Cal. He rose rapidly in the line of work he had chosen and held responsible positions with the United Gas & Power Co., the Exposed Treasure Mine and the Sierra Oil Co. In 1903 he became chief engineer of the Los Angeles Gas & Electric Corporation and later chief constructor. He was with this company for ten years when he became chief engineer and mechanical superintendent of the Alexandria Hotel in Los Angeles. In 1920 he was advanced to the position of supervising engineer of the Ambassador Hotel Corporation, a position which required a man of genuine ability so extensive was the system. Mr. Thomson continued in its upbuilding until it became the largest steam-generating plant west of Chicago. He held this position until late in 1922 when he resigned to take a rest before assuming the duties of mechanical superintendent of the new Biltmore Hotel of Los Angeles, one of a chain of the Bowman group of hotels.

Mr. Thomson became a member of the Society in 1921. He was a past-president of the National Association of Stationary Engineers.

DANIEL W. TOWER

Daniel W. Tower, president of the Grand Rapids Brass Co., Grand Rapids, Mich., died on February 12, 1923, failing to rally from an operation. Mr. Tower was born on August 15, 1855, in Oakfield, Mich. Leaving school at an early age, he entered the employ of the Phœnix Furniture Co., Grand Rapids, Mich., as an apprentice.

Three years later he became a draftsman with the firm of Nelson Mather & Co., where he remained for seven years. At the end of that period he became foreman of the Michigan Tool Works and in 1888 superintendent in charge of the design and construction of factory buildings and the layout and installation of power-plant equipment for the Works.

Mr. Tower was one of the founders of the brass industry in Grand Rapids, establishing a foundry which afterwards became the Grand Rapids Brass Co., of which he was president at the time of his death.

He became a junior member of the Society in 1900 and was advanced to full membership in 1907. He was prominent in local circles, an active member of the Grand Rapids Historical and of the Old Settlers Societies, and a Knight Templar of the masonic order. He was well known and beloved for his lectures and talks on his many journeys to distant lands as well as to the odd corners of his own country.

GEORGE A. WEBER

George A. Weber, an associate member of our Society since 1896, and formerly vice-president of the Rail Joint Co., Stamford, Conn., died on March 29, 1923. Mr. Weber was born on December 13, 1848, in Como, Ill., at that time a frontier town. He was educated in the public schools of Chicago and at Williston Seminary in Easthampton, Mass., from which he was graduated in 1870.

Mr. Weber was a pioneer in the successful development of base-supporting rail joints, and was the inventor of the Weber rail joint and other track appliances. In 1889 the Weber Rail Joint Co. was founded and incorporated in the State of West Virginia. In 1905 this company was merged with others to form the Rail Joint Co. Besides proving a most successful mechanical appliance, the Weber insulated joint is used throughoutAmerica by all railroads employing electrical signals and is also used extensively abroad.

Mr. Weber was also a member of the American Society of Civil Engineers, the Union League Club of New York, the New York Yacht Club, the Laurentian Club of Canada, and to a number of other organizations and clubs.

PETER WEBER

Peter Weber, who, at the time of his retirement in 1918, was president of the Sloan & Chace Manufacturing Co., Newark, N. J., died on May 9, 1923. Mr. Weber was born on May 21, 1860, in Lautersheim, Bavaria, Germany, where he was educated in the public and trade schools. He served a three-year apprenticeship in Germany and then came to this country at the age of sixteen.

He worked in various machine shops in and about New York City and finally in 1884 was employed by Thomas A. Edison, whose business at that time was located in Gerk Street, New York. Mr. Weber remained there until the Edison General Electric Co. acquired their plant at Schenectady, N. Y., and at that time he was transferred to those works as a machinist and tool maker. He advanced rapidly through the positions of inspector, assistant tool-room foreman, foreman and assistant superintendent to that of general superintendent in 1896. He retained that position until 1899 when he resigned to become general superintendent of the Edison Phonograph Works at West Orange. In 1912 Mr. Weber acquired the ownership of the Sloan & Chace Manufacturing Co., which plant he operated until late in 1918 when he was forced to resign because of ill health.

Mr. Weber became a member of the Society in 1913. He belonged to the masonic and Elks orders and was a member of the Edison Pioneers.

ARTHUR GORDON WEBSTER

Arthur G. Webster, one of the best known physicists and chemists in this country, and a member of the faculty of Clark University, Worcester, Mass., died on May 15, 1923. Professor Webster was born on November 28, 1863, in Brookline, Mass. He was graduated from Harvard University in 1885 and later studied in Berlin, Paris and Stockholm, receiving the degree of doctor of philosophy from the University of Berlin in 1890. He taught for a few years in Harvard and then became associated with Clark University as a professor of physics.

Professor Webster became a member of the Society in 1919. He was a fellow of the American Academy of Arts and Sciences, the American Philosophical Society, the American Institute of Electrical Engineers; an honorary member of the Royal Institute of Great Britain and a life member of the French Institute in the United States. He was a delegate of the U.S. Government to the International Radiotelegraphic Conference at London in 1912. He was a member of the committee on chemistry and physics of the U. S. Naval Consulting Board in 1915 and of the National Research Council in 1917. He belonged to the Phi Beta Kappa Fraternity and was a member of a number of clubs.

Professor Webster was the author of many scientific works upon the theory of electricity and magnetism and was the recipient in 1895 of the Elihu Thompson Prize of 500 francs for the best treatise on electricity.

SCHUYLER SKAATS WHEELER

A leading inventor and engineer in the electrical field, Dr. Schuyler Skaats Wheeler died suddenly of angina pectoris on April 20, 1923, in the sixty-third year of his age. Dr. Wheeler was born in New York and was graduated from Columbia University in 1881. After a few minor connections he became a member of the staff of Thomas A. Edison, and for three years was engaged in installation work. From 1886 to 1888 he was connected with the C. & C. Motor Co., which he helped to organize.

In 1888, the Crocker-Wheeler Company, of New York and Ampere, N. J., was formed with Dr. Wheeler as president, a position which he occupied up to the time of his death. This company was the first in the world to manu-

facture small electric motors, and during the past thirty-five years has installed more than 1000 electric drives designed by Dr. Wheeler.

Dr. Wheeler was an organizer and founder of the United Engineering Societies, and was a member of the national, civil, electrical, and mechanical engineering societies. He was for a time president of the American Institute of Electrical Engineers, to which he gave the Latimer-Clark library, said to be the largest collection of rare electrical books in existence. He became a member of the Society in 1899.

HENRY E. WHITAKER

Henry E. Whitaker, who, until his retirement in 1922, was mechanical engineer for Parke, Davis & Co., Detroit, Mich., died on June 10, 1923. Mr. Whitaker was born on October 27, 1858, in Forest City, Ill. He was educated in the schools of that city and attended the University of Michigan from which he was graduated in 1888 with the degree of B.S. in civil engineering.

From 1888 to 1894 he was in the employ of Jesse M. Smith, consulting engineer of Detroit. At the end of that time he became associated with F. A. Hubel of Detroit as a designer of special machinery. In 1901 he resigned from this position to become superintendent of the Pacific Iron Works of Seattle, Wash. Two years later he became assistant division superintendent of Parke, Davis & Co., Detroit, and in 1906 was advanced to the position of mechanical engineer, from which ill health forced him to resign in 1922.

Mr. Whitaker became a junior member of the Society in 1891 and was promoted to full membership in 1904. He was also a member of the Detroit Engineering Society from the early days of its organization.

CHARLES SUMNER WILLIAMSON

Charles Sumner Williamson, a member of the Society since 1914 and vice-president of the Mead-Morrison Manufacturing Co., Chicago, Ill., died on March 31, 1923. Mr. Williamson was born in April, 1874, in Cleveland, Ohio, where he received his early education. Later he attended the Case School of Applied Science from which he was graduated in 1895 with the degree of B.S. and three years later with the degree of C.E.

From 1894 to 1895 he worked in a civil engineer's office in Cleveland and at the end of that period became connected with the Brown Hoisting Machinery Co. of that city. From 1896 to 1898 he was in the employ of the King Bridge Co. and the Variety Iron Works, also of Cleveland. He resigned from his position with the latter concern to return as erection engineer to the Brown Hoisting Machinery Co., where he was located until 1904 when he became contracting engineer for Heyl & Patterson, Inc., Pittsburgh, Pa., in charge of the grab-bucket machinery department. Seven years later Mr. Williamson became associated as western manager and contracting engineer with the Mead-Morrison Manufacturing Co., Chicago, Ill., engineers and contractors for hoisting machinery. In 1918 he became vice-president of that concern.

HENRY CLINTON WILSON

Henry C. Wilson, consulting engineer, New York City, died on December 14, 1922. Mr. Wilson was born in June, 1876, in Washington, D. C. He attended Columbia University for four years, taking the course in electrical engineering, and then left in 1898 to enter the U. S. Army as a first lieutenant and later a captain in the Engineers Corps.

From 1901 to 1903 Mr. Wilson was in the employ of the Compound Beating Co., as engineer and designer, resigning to become associated with

the Sub-Target Gun Co., as chief and consulting engineer. Later he served as consulting engineer and technical director in the ordnance department of the Westinghouse Electric & Manufacturing Co. During the War Mr. Wilson was a major in the Coast Artillery Corps of the U. S. Army. Upon his return to civil life he entered the consulting field, in which work he was engaged at the time of his death.

Mr. Wilson became a member of the Society in 1915. He was also a member of the Society of Automotive Engineers, an associate member of the American Institute of Electrical Engineers and an associate of the U. S. Naval Institute.

HENRY EDGAR WOOD

Henry E. Wood, founder and general manager of the Wood Printing Co., Newark, N. J., died on February 24, 1923. Mr. Wood was born on June 16, 1870, in Fulton, N. Y. He was educated in the schools of St. Charles, Mich., and later took a course in mechanical drafting in the International Correspondence School. He served his apprenticeship in the Diltz Machine Shop, Oswego, N. Y., from 1888 to 1891. From 1891 to 1903 he worked as a general mechanic with the following firms: Dexter Folder Co., Fulton, N. Y.; Valley Machine Co., Saginaw, Mich.; F. W. Wheeler Co., West Bay City, Mich.; Industrial Machine Works, Bay City, Mich.; Russell & Co., Massillon, Ohio, and the Oswego Machine Works, Oswego, N. Y.

From 1903 to 1906 Mr. Wood was the manager and designer of paper-cutting machines with the Dexter Folder Co. in their Pearl River, N. Y., plant. For the next three years he was associated with the Auto Refrigerating Co., Hartford, Conn., as general foreman and shop manager, resigning to become general superintendent of the Tower & Lyon Co., Bloomfield, N. J. From 1910 to 1912 he served as general machinist with the General Electric Co., Harrison, N. J. Six years later he became connected with the Crocker-Wheeler Co., as an instructor of special mechanical workers. Later he founded the Wood Printing Co., Newark, N. J., of which he was general manager at the time of his death.

Mr. Wood invented a number of paper-cutting and labor-saving devices. He was the originator of Practical Don'ts for Machinists, Draftsmen and Electricians and was a frequent contributor to the technical press. He became an associate member of the Society in 1918.

H. S. HAINES

H. S. Haines, member of the Society since 1893, died at his home in Lenox, Mass., Nov. 3, 1923. Mr. Haines was born in Nantucket, Mass., Nov. 21, 1836. He served an apprenticeship with the Wilmington & Manchester R. R. Co., Wilmington, Del., and later obtained drawing-room and shop experience with the same railroad. He was engaged in civil engineering in railroad construction and as locomotive engineer and general superintendent for the N. & W. R. R. in Georgia, later serving as general manager and vice-president of that railroad.

Mr. Haines was for some time engaged in consulting engineering in New York and for Crivelli & Co., Switzerland, and spent several years previous to his death in England and in Springfield, Mass. He served as manager of the Society from 1896 to 1899.

LEBBEUS B. MILLER

Lebbeus B. Miller, who was born in Union Township, N. J., August 2, 1833, died July 23, 1923. Mr. Miller attended a private school in Elizabeth, N. J., and at the age of sixteen became apprenticed to E. & S. D. Gould of Newark, N. J., manufacturers of light machinery. In 1861 he became

associated with the Manhattan Firearms Co. of Newark, and in less than a year was made superintendent of the branch shops in Newark. In 1863 he was engaged by I. M. Singer & Co to design and supervise the construction of special tools for the production of the interchangeable parts of Singer sewing machines. He became superintendent of the factory of the Singer Mfg. Co. at Elizabeth, N. J., in 1869, and held that position until he retired from business in 1907.

Mr. Miller became a member of the Society in 1883 and served as a manager from 1893 to 1896. He was one of the founders and was the first president of the Elizabeth General Hospital.

ANTONIO C. PESSANO

Antonio C. Pessano, chairman of the board of directors of the Great Lakes Engineering Works, died at the Plaza Hotel, New York City, December 5, 1923. He was born in Philadelphia in 1857 and received his education by private instruction and at the Franklin Institute. For five years, terminating when he was twenty-one, he served an apprenticeship with Cresson and Smith. He then entered the drawing room of this company, working up to assistant superintendent of works. In 1884 he took full charge as designing and construction engineer of the George V. Cresson Company, later becoming vice-president and general manager of the company.

His entrance into the shipbuilding business was made in 1902 when the Great Lakes Engineering Works was organized to take over the business of S. F. Hodge and Company of Detroit. Mr. Pessano became president and general manager of the new organization which soon became a leading factor in shipyard work. During the war they supplied the United States Shipping Board with a number of steel ships. European shipping interests, also, have been supplied by their shipyards at Detroit and Ashtabula. Since 1920 Mr. Pessano has been chairman of the board of directors of the reorganized company. He was a member of our Society, of the Society of Naval Architects and Marine Engineers, a director of the American Merchant Marines Association, and a member of numerous other civic and professional associations.

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